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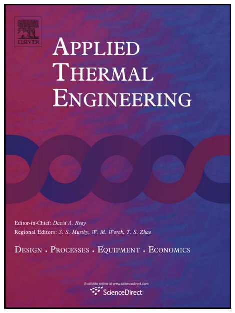

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Investigation on heat transfer evaluation for a more efficient two-zone combustion model in the case of natural gas SI engines

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Abstract

Two-zone model is one of the most interesting engine simulation tools, especially for SI engines. However, the pertinence of the simulation depends on the accuracy of the heat transfer model. In fact, an important part of the fuel energy is transformed to heat loss from the chamber walls. Also, knock appearance is closely related to heat exchange. 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, in the literature. The current study aims to investigate the effect of the choice of both the heat transfer correlation and burned zone heat transfer area calculation method and provide an optimized choice for a more efficient two-zone thermodynamic model, in the case of natural gas SI engines. For this purpose, a computer simulation is developed. Experimental measurements are carried out for comparison and validation. The effect of correlation choice has been first studied. The most known correlations have been tested and compared. Our experimental pressure results, supported for more general and reliable conclusions, by a literature survey of many other studies, based on measured heat transfer rates for several SI engines, are used for correlation selection. It is found that Hohenberg's correlation is the best choice. However, the influence of the burned zone heat transfer area calculation method is negligible.

Keywords: Heat transfer; Two-zone model; Natural gas; SI engines.

1. Introduction

Natural gas is one of the most interesting and promising available fuels for internal combustion engines. It has been recently used as an alternative to conventional fuels in order to satisfy some environmental and economical concerns. Moreover, 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.

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 to provide some qualitative trends.

Furthermore, two-zone model can represent a very interesting simulation tool especially for SI engines, due to the combustion type in this case. In fact, the flame front separates the chamber into a burned hot zone and a much cooler unburned zone.

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 [3]. 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, Ibrahim and co-workers [1,5] used Woschni correlation, Caillol and co-workers [6] used Hohenberg correlation and Soylu and co-workers [2,7], used Annand's. Also for heat exchange area estimation for the burned and unburned gases, several methods have been used. However, no justification for the choice has been given in the literature. Thus, this study aims to investigate the effect of the choice of both the heat transfer correlation and area calculation and provide an optimized choice for a more convenient 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.

2. Numerical model description

2.1 Model assumptions

The following assumptions and approximations are considered in the present work:

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.

   Thermal properties are determined by assuming ideal gas behavior for air-natural gas mixtures. Temperature variation is taken into account by using NASA polynomial expressions for each gas [3].
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) [1,8]. The temperature's variations of inner cylinder surface during the thermodynamic cycle are weak compared to the temperature's variations of the combustion gases [9]. In fact, the temperature of the wall can be considered as constant according to the results of Rakapoulos et al. [10].
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.

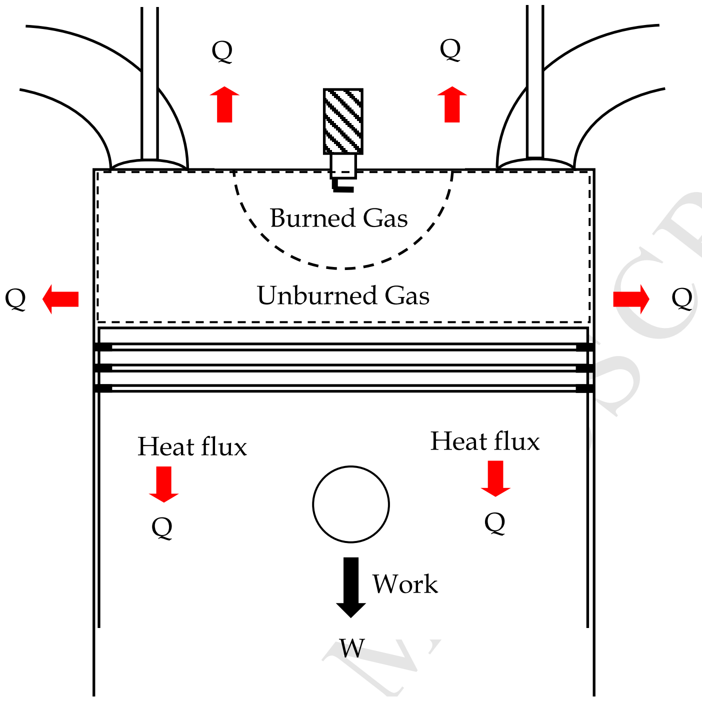


Fig. 1. Schematic representation of the two-zone combustion modeling

2.2 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, 11,12].

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.

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

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

Where $\sum_i$ is the summation of the heat transfer rates through the different engine's parts surfaces in contact with the cylinder gases.

Combustion sub-model

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 [12].

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

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-model (a review)

Heat transfers inside internal combustion engines are convective and radiative nature. However, for SI engines, the radiative transfers are negligible since they account for only 3 to 4% of the total heat transfer [8]. This cannot be applicable to diesel engines where the radiative transfers can represent up to 10 % of the heat exchanges due to soot formation during combustion [8].

During combustion, the burned gas temperature increases significantly with maximum which can reach about 2800 K. This induces gases expansion and thus an increase in their movement. It is during this period that heat transfers are the most important. Heat flux induced can reach several tens megawatts per square meter for some engines [3].

These heat transfers between gases and the chamber walls are non-uniform and unsteady, hence generally complex to evaluate. Many approaches can be used, depending on the kind of results required.

However, it is frequent to simplify the formulation, and use the Newton relation (8). It is the approach adopted in zero-dimensional models.

$$\dot{Q}_w = h_g \cdot S_w \cdot (T_g - T_w) \quad (8)$$

where T_g is the gases temperature, T_w the wall temperature and h_g the gas-wall heat transfer coefficient . Like T_g, h_g is supposed to be uniform in all the parts of the chamber with same gas (burned or unburned).

Accordingly, the main parameter to determine, in order to evaluate the parietal losses during an engine cycle, is the heat transfer coefficient h_g.

A wealth of literature has been published over the years regarding the gas-to-wall heat transfer process in SI and CI engines and a number of correlations have been proposed for calculating the instantaneous heat transfer coefficient [13-20].

These correlations provide a heat transfer coefficient representing a spatially-averaged value for the cylinder. Therefore, they are commonly referred to as global heat transfer models [13].

The most known ones were inventoried by Trapy [19], Borman and Nishiwaki [20], Guibert [3] and Ollivier [8].

Those correlations can be classified into two categories, according to the assumption retained for the heat transfer origin. However, we will detail only the ones which will be examined.

Natural convection assumption

The first correlations established for the heat transfer evaluation in engines, adopted the assumption of natural convection. The heat transfer coefficient, is then, written, in a dimensionless form, according to the relation:

$$Nu = Cste \cdot (Gr \cdot Pr)^n \tag{9}$$

The first model dates back to 1923 and was established by Nüsselt. It consists of an empirical correlation established by tests carried out in spherical bomb. This model was followed and adjusted by Brilling , and Eichelberg; by using the experimental test results carried out on internal combustion engines [3].

These models had the advantage of describing, for the first time, the influence of engine parameters such as gas temperature, pressure and engine speed. But this approach has quickly reached its limits. Woschni [16] explains, furthermore, why these formulas are finally only very approximate. This approach was thus discussed until being abandoned in favor of the use of dimensional analysis considering forced convection.

However, Eichelberg's correlation, even though categorized in this family [8] still gives good estimation for heat transfer calculation.

- Eichelberg's Correlation (1939) [14]

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

Forced convection assumption

Because of the inadequacy of the natural convection assumption, correlations based on this assumption apply with difficulty to other engines. Thus, some experimenters (Annand, Woschni, Hohenberg, …) adopted the forced convection assumption [8]. This one is more realistic because the fluid movements in the chamber are the consequence of external mechanical actions.

These studies have generally relied on dimensional analysis for turbulent flow that correlates the Nusselt, Reynolds, and Prandtl numbers. Using experiments in engines and applying the assumption of quasi-steady conditions has led to empirical correlations for both SI and CI engine heat transfer.

The general formulation of the Nusselt number, considering some assumptions, is then written:

$$Nu = a \cdot Re^m \cdot Pr^n \tag{11}$$

Substituting Nu and Re with physical properties, the global heat transfer coefficient depends on characteristic length, transport properties, pressure, temperature, and characteristic velocity. A scaling factor is used for tuning of the coefficient to match specific engine geometry. A value for the exponent *m* has been proposed by several authors, for example, *m* = 0.5 for Elser and Oguri, 0.7 for Annand and Sitkei, 0.75 for Taylor and Toong, and 0.8 for Woschni and Hohenberg. Except for the Woschni's correlation, most of these correlations use a time-averaged gas velocity proportional to the mean piston speed. However, Woschni separated 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.

The Woschni's and Hohenberg's correlations are the most known of this category.

- Woschni's Correlation (1965-68) [16]

$$h_g = C_0 \cdot \left[B^{-0.2} \cdot P^{0.8} \cdot \left((C_1 \cdot V_{mp}) + C_2 \cdot \frac{V_d \cdot T_1}{P_1 \cdot V_1} \cdot (P - P_{mot}) \right)^{0.8} \cdot T^{-0.53} \right] \tag{12}$$

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's Correlation (1979) [18]

Hohenberg [18] noted that the Woschni correlation underestimates the heat transfer coefficient during compression and over-estimates it during combustion. This leads to an over-estimate of the average heat flux during a cycle. Moreover, he underlines its difficulty of use. This leads him to propose the following correlation

$$h_g = C_1 \cdot C_u^{-0.06} \cdot P^{0.8} \cdot T^{-0.4} (C_2 + V_{mp})^{0.8} \tag{13}$$

P is the instantaneous pressure, in bar. The numerical values C_1=130 and C_2=1.4, appearing in equation (13), are constants established on base of six representative engines.

- Sitkei's Correlation [17]

This correlation belongs to the same family as Woschni and Hohenberg correlations. This correlation is also established on base of Diesel engine experiments. It is expressed according to the equation (14):

$$h_g = 2.36 \times 10^{-4} \cdot (1+b) \cdot \frac{(P \cdot V_{mp})^{0.7} A^{0.3}}{T^{0.2} \cdot (4V)^{0.3}} \tag{14}$$

With b = 0 - 0.35

Specific correlations for SI engines

The correlations given previously are typically established on base of diesel engines experiments. Consequently, they are theoretically not suitable to model the in-cylinder heat transfer process for SI engines, because their combustion principles and operating ranges are very different. However, many studies used those correlations for SI engines. For instance, Oguri [21] used Eichelberg's model to predict the heat transfer rate of a 1400 cm^3 SI engine, yielding predicted results that agreed with the experimental results for the expansion stroke, but not for the compression stroke. Shayler [22] calculated heat transfer using Woschni's, Annand's, and Eichelberg's experimental models. It was found that Eichelberg's model could produce prediction results closest to the experimental data.

Correlations were established especially for this type of engine, but they are fewer. Trapy's Correlation [19] is one belonging to this category. Unlike those defined previously, this correlation does not have a universal character and the constant coefficients appearing in it, differ from an engine to another.

Finally, Annand's correlation has been established for both SI and CI engines, the constant *b* differs depending on the type of engine.

- Annand's Correlation [15]

$$h_g = a \cdot \frac{k_g}{B} \cdot \mathrm{Re}^{0.7} + b \cdot \frac{(T_g^4 - T_w^4)}{(T_g - T_w)} \tag{15}$$

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

Note that when the expression of the previous correlations is different depending on the engine cycle phase, the appropriate formula is specified. Otherwise, the expression is unique

2.3 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, at inlet valve opening, the thermodynamic cycle simulation starts with assumed guesses of the values of pressure and temperature of the contents within the cylinder. Those values are used in order to estimate the initial value for the mass of the content within the cylinder, via the ideal gases equation applied to the burned gas mass. At each crank angle, the flow rate of inlet fresh gases and exhaust gases are calculated. Hence, the mixture within the cylinder is well defined for each angle. After two crankshaft revolutions, 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 [1,2,5]. Moreover, experimental data are used for model validation.

3. Experimental setup description

3.1. Description of facilities

For comparison and validation, experimental data are used. The experimental setup consists of a single cylinder Lister Peter direct injection Diesel engine, adapted for gas carburetion (Fig. 2). The combustion chamber is a bowl-in-piston type (Fig. 3).

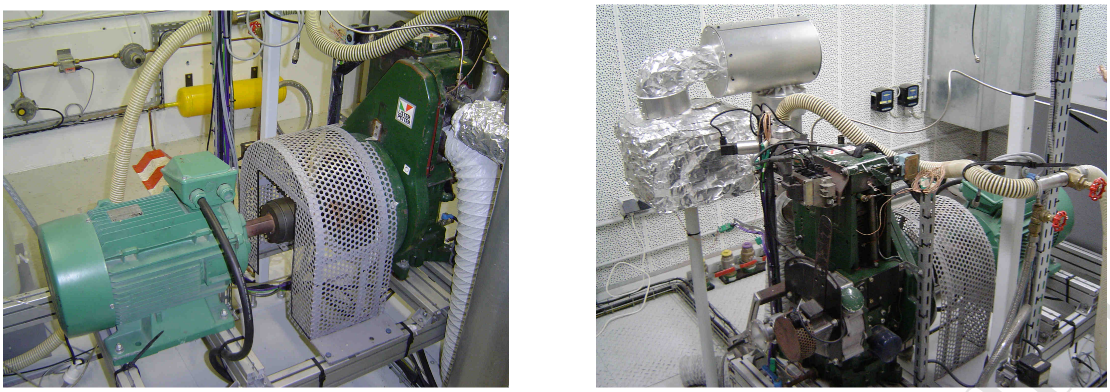


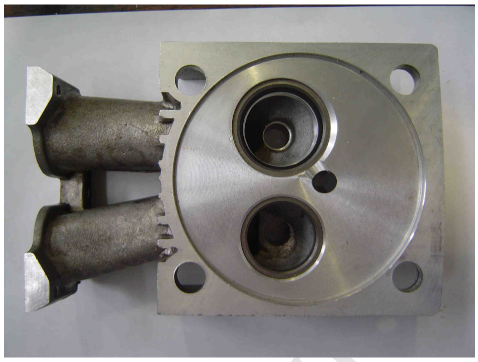
Fig. 2. Lister-Petter Bench engine

The engine specifications and bowl dimensions are provided in Table 2. The engine is naturally aspired. Tests were conducted at a constant speed (1500 rpm).

Table 2: Engine Specifications

Constructor	LISTER-PETER
Engine type	Four strokes Spark ignition
Number of cylinders	Single cylinder
Cooling	Air cooled
Bore x stroke	95.5 x 88.7 mm
Volumetric capacity	635 cm^3
Dead volume	53 cm^3
Compression ratio	12.98 : 1
Connecting road length	165.3 mm
IVO	44° c.a. before TDC
IVC	71° c.a after BDC
EVO	86° c.a. before BDC
EVC	58° c.a after TDC
Inlet valve diameter	45.4 mm
Inlet valve max lift	10 mm
Exhaust valve diameter	37.8 mm
Exhaust valve max lift	10 mm
Bowl diameter	45 mm
Bowl depth	15 mm
Bowl eccentricity	6.25 mm

(*a*)



(*b*)

Fig. 3. Illustration of the geometrical configuration of the piston (*a*) and the cylinder head (*b*).

3.2. Experimental data acquisition system

Two acquisition modes are used (Fig. 4):

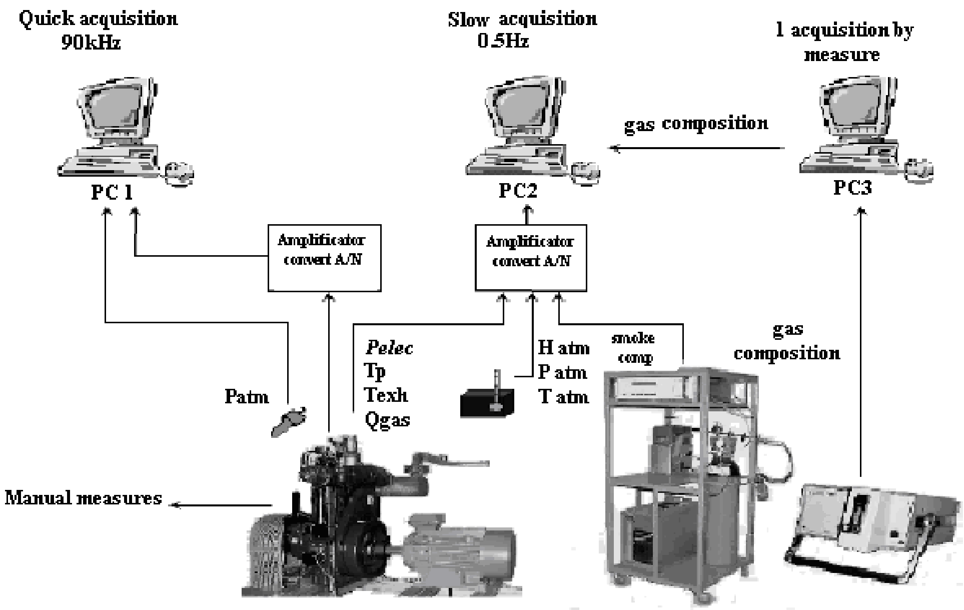


Fig. 4. Experimental measures acquisition system synoptic

- Slow acquisition (0.5 Hz): The data concerned by this type of acquisition are the air and fuel flow rates.

Natural gas flow rate is measured using a thermal mass flowmeter (Instrutec F112AC-HB-55V). This type of flowmeter allows 1% precision measure. Flow rate measure takes into consideration the gas composition. Like for the burned gas flow, the air intake is in pulsated mode. The existence of this mode requires the use of a buffer volume in order to deaden the flow pulsations. Consequently, the use of an orifice flowmeter becomes possible to measure the air flow. This one is made of a diaphragm of 70 mm diameter installed at the buffer volume inlet.

The equivalent air fuel ratio is calculated in two ways: either from the gas to air ratio, or from the exhaust gas analysis.

- Quick acquisition (90 kHz): This type of acquisition is used for the intake and in cylinder pressures. It is also used for the crank angle measures.

The in-cylinder pressure is measured using a piezoelectric sensor AVL QH32D installed on the cylinder head. The range of this sensor is 0-200 bar. The crank angle determination is realized with an angular encoder (AVL 364C), installed on the crankshaft. This encoder type allows 0.1 ° CA resolution. This acquisition management is ensured by *Indiwin* software of AVL. The intake pressure is measured using a piezoresistive sensor 0-2.5 bar, installed in the intake port.

The capacitive TDC-Sensor (AVL 428) is used to determine TDC dynamically in a motored engine. The TDC position is determined with crank angle accuracy less than 0.1° CA [23].

4. Study purposes and strategy

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, in the literature. The current study aims to investigate the effect of the choice of both the heat transfer correlation and burned zone heat transfer area calculation method and provide an optimized choice for a more efficient two-zone thermodynamic model, in the case of natural gas SI engines.

Firstly, in order to investigate the importance of the heat transfer correlation choice, the influence of the correlation selection on the calculated performances is studied. After that, the influence of the correlation accuracy in the pertinence of a two-zone model, by selection of realistic Wiebe function parameters is highlighted. Lastly, in order to provide an optimized correlation choice for SI engines, our experimental results are used to explore for an optimized choice of the heat transfer correlation. The results showed a same tendency for all the cases investigated despite the limited experimental setup. The engine used for comparison is just one at a single rotation speed. Therefore, in order to get more general and reliable conclusions, a literature survey of many other studies [18,22,24], which were based on measured heat transfer rates for different SI engines, is conducted. Their results are in concordance with our experimental results. Hence, conclusions based on our experimental results and on a literature investigation are deduced.

Finally, the influence of the heat transfer area calculation method is investigated. Two, practical methods are examined.

5. Results and discussion

5.1 Heat transfer correlation choice

a. Influence of the heat transfer correlation choice on calculated performances

In order to show the effect of the heat transfer correlation on the calculated engine's performances, two cases are exhibited (Fig. 5 a, b).

To highlight this influence, the main parameter used, is the cylinder pressure. This choice is not fortuitous. It is justified by two reasons. First, it is one of the main outputs of a two- zone model, which is the calculation tool under consideration. Also, it is the calculation base for several other engine performances. A mean pressure cycle was obtained by averaging 100 cycles acquired in sequence [25].

Moreover, a good prediction of the cylinder pressure during combustion can be used to detect the risk of the knock phenomenon. The in cylinder temperature can also be used for this purpose. Knock is an abnormal combustion in the cylinder of spark ignition (SI) engines. It is the result of autoignition of one part of the end gas because of the rise in temperature and pressure due to propagation of the primary flame front [26]. This phenomenon is very undesirable because it results in lower engine efficiency, an increase of some emissions and even leads to destruction of the engine under heavy knock operation.

Thus, It appears clearly (Fig. 5, 6, 7), 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 (Fig. 8, 9). The details are given while discussing each correlation alone, hereafter.

Also, if the engine is subject to high mechanical and thermal loads, it can lead to its damage. Prediction of such risks is of a great importance. Predicted maximum pressure and maximum temperature can be used for this objective. However, the cases exhibited (Fig. 6, 7) show visibly the influence of the correlation choice on the predicted Maximum pressure and maximum temperature. In fact, the difference in the maximum pressure exceeds 5 bars, and is around 100 °C for the maximum temperature.

Even though, it's well known that 2 zone models cannot pretend for big prediction accuracy, but they can be a good first investigation tool, if well constructed. However, if the above mentioned differences are added to the inaccuracies produced by the model approximations, this will diminish considerably the model value.

b. Influence of the correlation accuracy on the two-zone model pertinence

Two-zone model is one of the experimentally based simulation tools. Generally, for SI engines, Wiebe function (Eq. 7) is used for heat release evaluation. In this function, a_w and m_w are adjustable parameters which fix the shape of the curve [12]. The parameter a_w interprets the burned fuel mass fraction compared to the introduced fuel mass. Considered 99.9% of the fuel introduced as burned, gives a value of 6.908 for a_w. The variation of the form factor m_w, involves a dissymmetry of the distribution, therefore of the heat release. A low value involves very violent combustion beginning. The increase in m generates late combustions and a shift of the combustion peak [11].

Model calibration implies choice of those parameters. The calibration is as more efficient as those parameters are realistic. The use of the calibrated model for new engines, in a predictive manner, to provide some qualitative trends, is then more efficient. However, those parameters are more realistic when the calibration is carried while using a more accurate heat transfer model. Hence, the two-zone model pertinence depends on the heat transfer correlation convenience.

c- Heat transfer correlation choice optimization

Correlation choice for an optimized two-zone model must take into consideration the results accuracy, but also its practical convenience and calculation time.

Our experimental pressure results are used to explore for an optimized choice of the heat transfer correlation.

Actually, the cylinder pressure prediction depends on the heat transfer and the heat release rates estimations, for of course, same engine design and operating conditions. However, for a given heat release rate, the cylinder pressure depends only on the heat transfer correlation choice.

For the investigated cases, in order to see the influence of the heat transfer correlation choice, the Wiebe function parameters are taken the same for all examined correlations. Moreover, to use realistic parameters, physical considerations and several previous studies [1,3,8] for similar cases are used for their selection. Also, they are the suitable values for the majority of the correlations. Naturally, a check and validation is carried out on the engine. The values used for the investigated cases are: $a_w = 2.1$, $m_w = 6.9$

Several operating conditions have been explored. The results showed a same tendency for all the cases studied even though with limited experimental setup. Two cases are exhibited (Fig. 5 a, b).

However, in order to provide more reliable and general conclusions, results of many other studies [18,22,24] in the literature, which were based on measured heat transfer rates for different SI engines are investigated. They are in concordance with our experimental results. They are given while discussing each correlation separately, hereafter.

- Woschni: Like noted by Hohenberg [18], this correlation underestimates the heat transfer coefficient during compression and over-estimates it during combustion. However, the effect on the cycle performance is negligible.

  This correlation separates the referenced 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. Its need to motoring pressure makes it difficult to use. It also needs more calculation time comparing to other correlations.
- Hohenberg: It gives the closest results to our experimental data. Besides, like mentioned before, Hohenberg [18] proposed this correlation, as an improvement, to mitigate some lacks noted on the Woschni correlation. Also, the deficiency of the Eichelberg correlation during compression is no more found with the results of this correlation (Fig. 8 a, b). Moreover, it is easy to use, and the calculation time is minimized. Almost no tuning has to be performed. It is our best choice.
- Eichelberg: Except for compression stroke (Fig. 8), as mentioned in reference [24], this correlation results are acceptable. Moreover, Shayler [22] calculated heat transfer using Woschni's, Annand's, and Eichelberg's experimental models for SI engines. It was found that Eichelberg's model could produce prediction results closest to the experimental data.

  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 in SI engines.

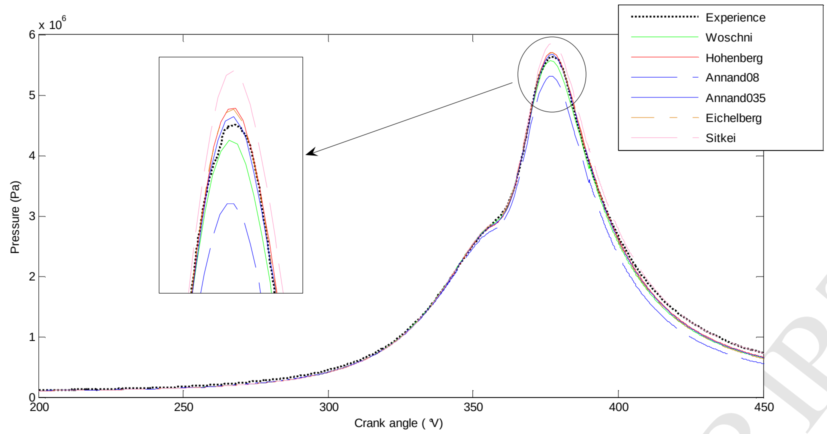
On the other hand, tuning coefficient has a big effect on the heat transfer coefficient and the corresponding engine cycle performance. In fact, if the (a) parameter is set to 0.8, the heat transfer coefficient is overestimated (Fig. 8) and hence the engine cycle performance is underestimated (Fig. 5, 6, 7). This makes it not very interesting to use in thermodynamic models.

- Sitkei: The heat transfer coefficient is underestimated (Fig. 8) and the engine cycle performance is consequently overestimated (Fig. 5, 6, 7). The accuracy in this case is not acceptable.

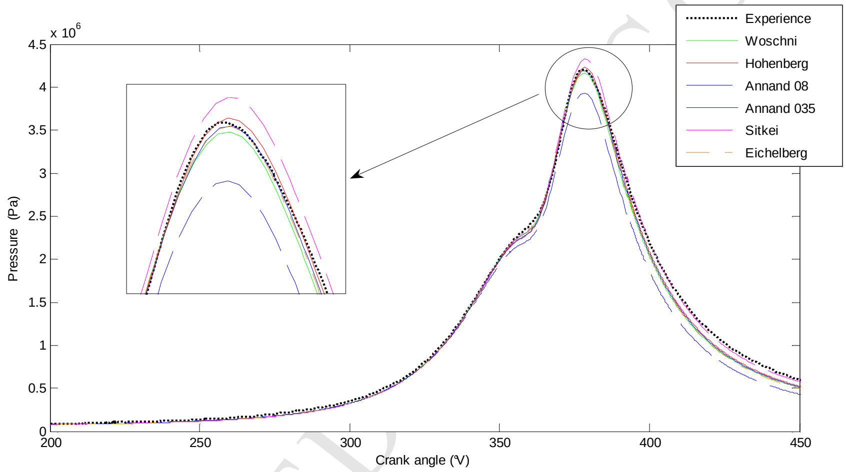
The following table summarizes the previous comparing elements. It is an interesting tool. It can be used as a guide for correlation choice, basing on several criteria, for SI engines.

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	Not acceptable	Easy	Good	Need

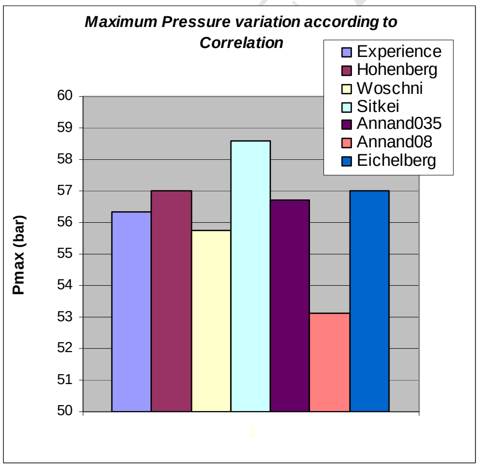


(a)

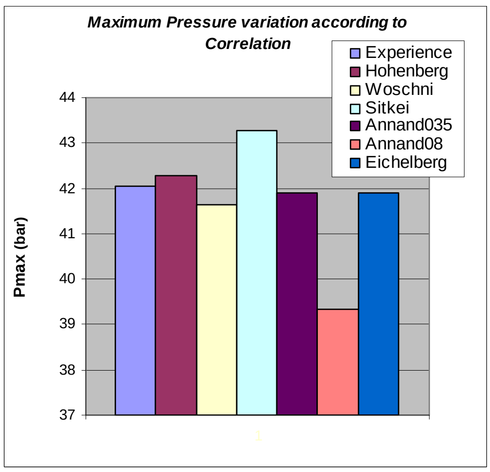


(b)

Fig. 5. 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*

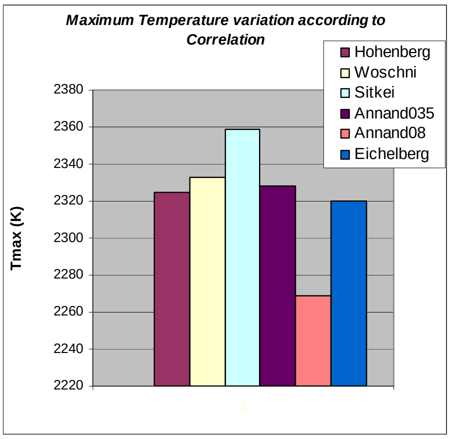


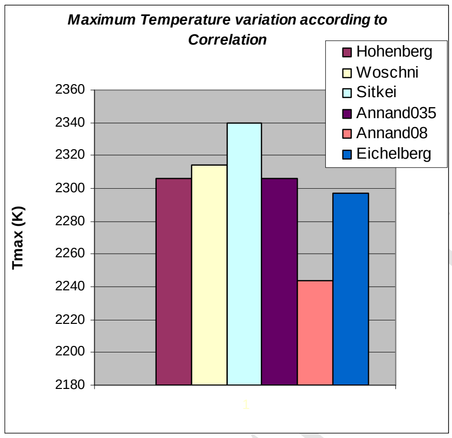
(a)



(b)

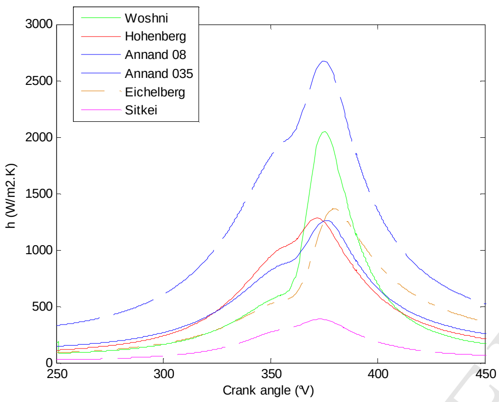
Fig. 6. Maximum Pressure comparison for different correlations within two cases:
(a) Ø=1,α=9° c.a, Full load *(b) Ø=1,α=9° c.a, Partial load*

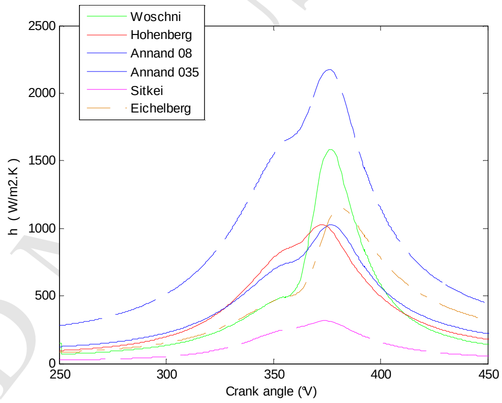




(a) (b)

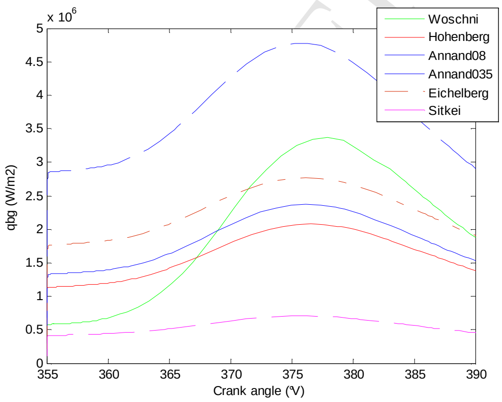
Fig. 7. Maximum temperature comparison for different correlations within two cases:
(a) Ø=1,α=9° c.a, Full load *(b) Ø=1,α=9° c.a, Partial load*

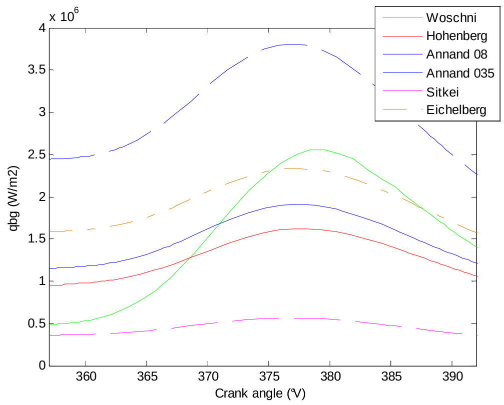




(a) (b)

Fig. 8. Average 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)

Fig. 9. 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*

5.2 Heat transfer area calculation method influence

Two calculation methods used in previous studies have been compared

- First method

The heat transfer rates for both zones were determined as if each of the zones filled the entire cylinder.

Then, the heat transfer rates were multiplied by the fraction of the cylinder volume that is occupied by the zone. This method of volume weighting was used by Shapiro and Van Gerpen [27] and by Soylu [2].

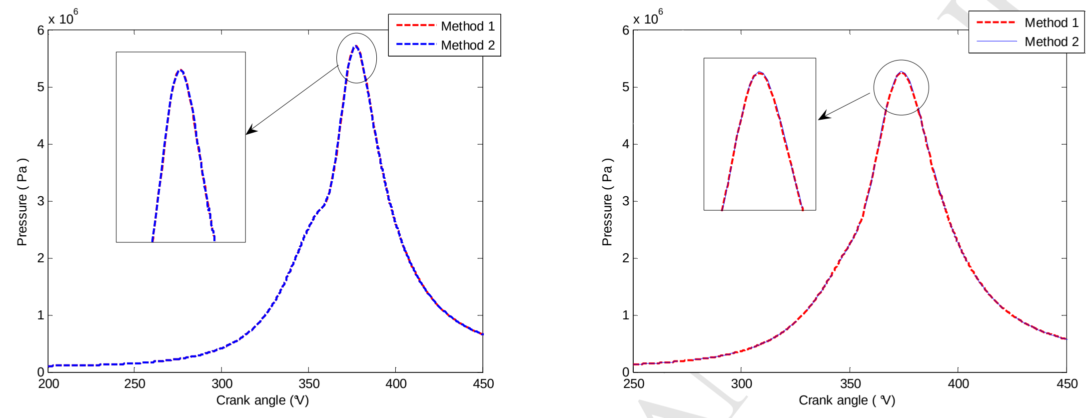
$$A_b = \frac{V_b}{V_u + V_b} \cdot A \qquad (16) \qquad\qquad A_u = \frac{V_u}{V_u + V_b} \cdot A \qquad (17)$$

- Second method

The combustion chamber wall area in contact with the burned gases is assumed to be proportional to the square root of the burned mass fraction to account for the greater volume filled by burned gases against the unburned volume as suggested by Ferguson [28]. This method is also used by Ibrahim and Bari [1].

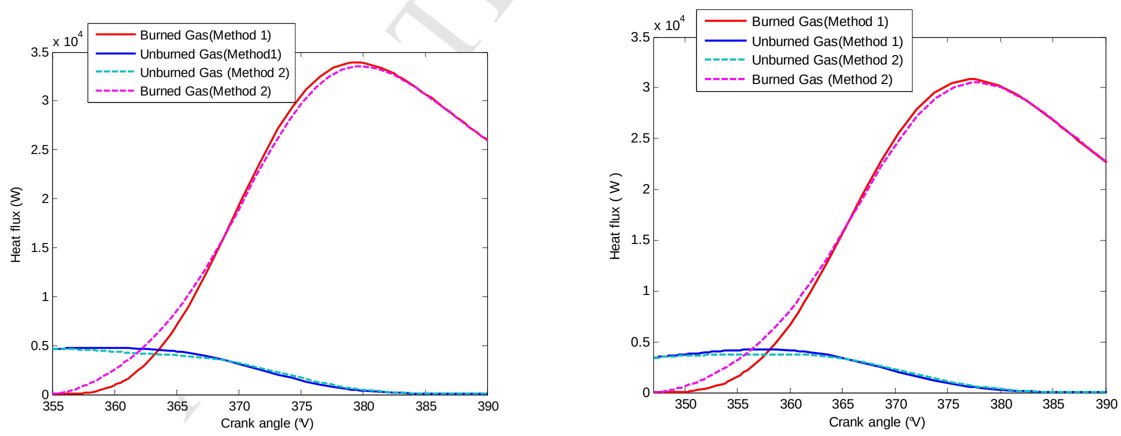
$$A_b = x_b^{1/2} \cdot A \qquad (18) \qquad\qquad A_u = (1 - x_b^{1/2}) \cdot A \qquad (19)$$

In order to compare the results of area estimation methods, different engine operating conditions have been considered. Two cases are exhibited (Fig. 10, 11). The heat transfer correlation used is Hohenberg's. The difference of heat flux is negligible (Fig. 11). Almost no effect of the estimation method on the engine cycle performance is distinguishable (Fig. 10)



(a) (b)

Fig. 10. Pressure comparison for different heat transfer area calculation methods within two cases:
(a) Ø=1, α=9° c.a, Full load *(b) Ø=0.85, α=17° c.a, Partial load*



(a) (b)

Fig. 11. Heat flux comparison for different heat transfer area calculation methods within two cases:
(a) Ø=1,α=9° c.a, Full load *(b) Ø=0.85,α=17° c.a, Partial load*

6. Conclusion

In previous works using two-zone model for natural gas SI engines, many heat transfer correlations and heat transfer area estimation methods have been used and no choice justification has been given.

In order to provide an optimized tool for future studies, using thermodynamic modeling for SI natural gas engines, the two aspects have been investigated in the present work. First, the effect of heat transfer correlation choice has been studied. The most known correlations have been tested and compared considering different aspects. Our experimental pressure results, supported for more general and reliable conclusions, by a literature survey of many other studies, based on measured heat transfer rates for several SI engines, are used for correlation selection. It is found that Hohenberg's correlation is the best choice. It gives the more accurate results. It is easy to use, and the calculation time is minimized. Almost no tuning has to be performed. Even though not like the first one, Eichelberg's correlation is also an acceptable choice.

For the heat transfer area calculation, two estimation methods, used in previous studies, have been tested. Even though the volume weighting method seems to be more logical, it is found that the influence of the method is negligible.

Notation

A	Instantaneous surface area exposed to heat transfer, [m^2]
B	Cylinder bore or diameter, [m]
Gr	Grasshof number, [-]
h	Enthalpy, [J.kg^{-1}]
h_g	Gas-wall heat transfer coefficient, [W/m^2 K]
k_g	Thermal conductivity of the gas, [W/m K]
m	Mass of the gas, [kg]
N	Engine rotational speed, [rpm]
Nu	Nusselt number, [-]

P Instantaneous cylinder gas pressure, [Pa]
Pr Prandtl number, [-]

q Heat flux rate (heat transfer per unit area), [W/m^2]
Q Exchanged Heat, [J]
Re Reynolds number, [-]
S_w The area of the surface in contact with the gases, [m^2]
t Time, [s]
T Temperature, [K]
T_{w} Wall surface temperature, [K]
u Internal energy, [J.kg^{-1}]
V Instantaneous cylinder volume, [m^3]
V_d Volumetric capacity, [m^3]
V_{mp} Mean piston speed, [m/s]
x_b Burned gas mass fraction, [-]

Greek Symbols

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

Subscript and Superscript

a Admission
b Burned
e Exhaust
mot Motoring
u Unburned

Abbreviations

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

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