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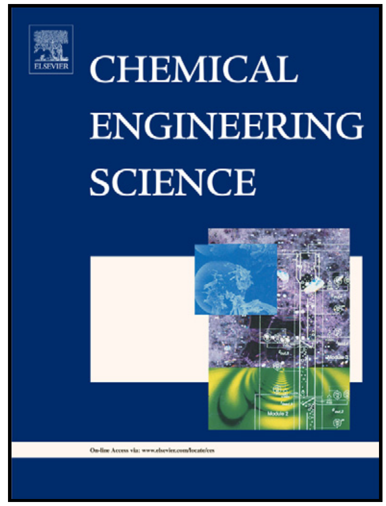

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Theoretical Limits of Scaling-Down Internal Combustion Engines

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Abstract

Small-scale energy conversion devices are being developed for a variety of applications; these include propulsion units for MAV (micro aerial vehicles). The high specific energy of hydrocarbon and hydrogen fuels, as compared to other energy storing means, like, batteries, elastic elements, flywheels and pneumatics, appears to be an important advantage, and favors the ICE as a candidate. In addition, the specific power (power per mass of unit) of the ICE seems to be much higher than that of other candidates.

However, micro ICE engines are not simply smaller versions of full-size engines. Physical processes such as combustion, and gas exchange, are performed in regimes different from those occur in full-size engines. Consequently, engine design principles are different at a fundamental level, and have to be re-considered before they are applied to micro-engines. When a Spark-Ignition (SI) cycle is considered, part of the energy that is released during combustion is used to heat-up the mixture in the quenching volume, and therefore the flame-zone temperature is lower and in some cases can theoretically fall below the self-sustained combustion temperature. The flame quenching thus seems to limit the minimum dimensions of a SI engine. This limit becomes irrelevant when a Homogeneous-Charge Compression-Ignition (HCCI) cycle is considered. In this case friction losses and charge leakage through the cylinder-piston gap become dominant, constrain the engine size, and impose minimum engine speed limits.

In the present work a phenomenological model has been developed to consider the relevant processes inside the cylinder of a Homogeneous-Charge Compression-Ignition (HCCI) engine. An approximated analytical solution is proposed to yield the lower possible limits of scaling-down HCCI cycle engines. We present a simple algebraic equation that shows the inter-relationships between the pertinent parameters, and constitutes the lower possible miniaturization limits of IC engines.

Nomenclature

a	Friction polynomial coefficient	kPa
a_c	Speed of sound at charge conditions	m/s
a_0	Speed of sound at ambient conditions	m/s
b	Friction polynomial coefficient	$kPa*min$
c	Friction polynomial coefficient	$kPa*min^2$
D	Cylinder bore diameter	m
FA	Fuel/air mass ratio	-
fp	Friction power	kJ/s
h	Apparent heat of combustion	kJ/kg
h_c	Heat of combustion	kJ/kg
$imep$	Indicated mean effective pressure	kPa
ip	Indicated power	kJ/s
L	Piston stroke	m
$m_{cylinder}$	Trapped mass	kg
$m_{cylinder\_max}$	Maximum trapped mass	kg
N	Engine speed	rev/min
r_v	Compression ratio	-
$T_{compression}$	Temperature after compression	K
T_{ig}	Ignition temperature	K
T_0	Ambient temperature	K
$tfmep$	Total friction mean effective pressure	kPa
$\overline{u}_{piston}$	Mean piston velocity	m/s
V	Displacement volume	m^3
Greek		
α	Mass loss fraction	-
$\Delta t_{charging}$	Charging time interval	s
δ	Cylinder-piston gap	m
η_{ch}	Charging efficiency	-
η_{th}	Cycle thermal efficiency	-
ρ_c	Charge density	kg/m^3
ρ_0	Ambient air density	kg/m^3
τ	Scavenge port open	s

Introduction

In recent years there has been an evident increase in micro-scale electrical and mechanical device development. From unmanned ground and air vehicles to hand held cellular phones, the emphasis is on size reduction and increased operating envelope[1,2,3]. The term micro aerial vehicle (MAV) refers to a smaller-size of remotely controlled aircraft (UAV). The target dimension for MAVs today is approximately 10-15cm, while development of insect-size aircrafts, though a highly-motivated challenge, is not expected in the near future. Potential military, scientific, police and mapping are the more important applications that motivate the target. Another promising area is remote observation of hazardous environments which are hard or in-accessible to ground vehicles. MEMS technology has brought major improvement in all aspects except one; power sources[2,3]. In order to comply with miniaturization, the need for high energy-density power units is becoming more and more apparent. Large scale high energy-density sources, such as jet engines and turbines are commonplace, yet no sources that suit the micro scale devices currently exist. Over the last two decades the energy density of chemical storage cells (batteries) has only increased by a factor of two (on average 600kJ/kg alkaline and 1,200kJ/kg lithium), far less than the increasing demand[5,6,7]. The growing gap between demand and supply has led intense search elsewhere. In these efforts, owing to the superior energy densities of hydrocarbon fuels (on average 30,000-40,000kJ/kg), it is immediately evident that herein lies a possible solution. Even with thermal efficiencies as low as 5%, ICE converted hydrocarbon fuels are still superior to battery-motor electrical systems[8].

Micro-Engines are not simply smaller versions of their macro-scale counterparts. Many of the major components cannot be easily reduced in order to achieve proper functioning and it is clear that the micro scale engine is to be very different from the macro scale known to us[6]. Physical processes such as combustion, heat transfer and gas exchange are to be re-evaluated at a fundamental level before they may be applied to micro-engines. Research has shown that a power law was found that relates engine power to engine mass[8]. Additional concerns are friction, sealing and fabrication limitations[6].

Lee and Kwon[4] suggested that lower limit of the combustor size and the combustion efficiency in small-scale combustors are two of the most prominent questions. A major problem in the development of a micro-engine is the combustion in a very limited volume. Existing understanding of combustion in a volume that is comparable to the laminar flame thickness is limited[9]. One of the impeding factors is the quenching distance. Quenching phenomena in micro scale combustion may be due to two primary mechanisms, thermal and radical quenching. Thermal quenching occurs when the heat loss through the chamber wall is too high for the combustion process to be self-sustained. Radical quenching occurs due to adsorption of radicals on the combustion chamber walls that results in lack of homogeneous combustion[2]. Both mechanisms become more significant in the micro-scale, the former being of more concern than the later. Common values of boundary flame quenching of hydrocarbon fuels are 1mm. Considering a combustion chamber that is as small as 4mm in diameter conveys the problem. How to prevent thermal quenching and achieve stable combustion is one of the most challenging issues in micro-combustion[10].

Another concern is residence time. In order to achieve stable combustion, reactant residence time must be longer than the chemical reaction time. On the macro-scale, residence time is always long enough and is never a limiting factor. Decreasing the combustion chamber's dimensions leads to significant reduction of residence time since the flow speed cannot be reduced accordingly[11]. In order to maintain a working cycle and sufficient power output, higher engine velocities must be maintained. This leads to shorter residence time while chemical reaction time stays the same. This phenomenon has subsequent consequences since insufficient residence time leads to incomplete or partial combustion, in turn leading to insufficient heat generation and further quenching[12].

Substantial heat transfer to the boundaries could influence micro-combustion. Heat generation is proportional to the combustor volume while heat loss is proportional to the surface area. This ratio is inversely proportional to the characteristic dimension of the combustion chamber. Thus, the heat transfer rate increases as the chamber dimension decreases. This is advantageous in micro-chemical systems because highly exothermic processes such as catalytic partial-oxidation reactions and the catalytic oxidation can be controlled[5]. In our case, however, this phenomenon is counter-productive since excessive heat loss will in turn lead to increased quenching and eventually to output losses. Reduced scale also increases the temperature gradient at the jacket and further increases the conduction heat loss.

The characteristic length of the combustion chamber and the reacting gas flow path in micro-combustors is still sufficiently larger than the molecular mean-free path of the air and other gasses flowing through the system. Hence, the fluid media can be reasonably considered as continuous[2]. Since all the characteristic dimensions in micro-combustion are small and Reynolds numbers are well under turbulent values, a laminar viscous flow may be considered. Sher and Levinzon[13] showed that even if perfect mixing, and infinite flame speed are considered for a spark-ignition engine, the energy released by the fuel strongly depend on the effective volume of the burning volume. This in turn, determines the maximum cycle temperature, and thus its efficiency and specific power.

Homogeneous Charge Compression Ignition (HCCI), was first identified and studied by Onishi[14]. It involves compressing a homogenous fuel-air mixture until it self-combusts. It differs from Diesel combustion because the charge is premixed and only then compressed and it differs from SI combustion because the ignition is spontaneous and not spark-induced. The combustion occurrence depends on chemical kinetics and the compression process[15]. Verified characteristics of HCCI include: 1. Ignition occurs simultaneously at numerous locations within the combustion chamber, 2. Traditional flame propagation advance is absent, 3. There is no need of an external event to initiate ignition, 4. Both rich and extremely lean fuel mixtures may be used, 5. Charge is consumed rapidly, 6. A wide variety of fuels may be used, 7. Essentially, no compression ratio limitations[5,6,16]. When considering micro-scale applications, HCCI advantages are an apparent answer to most of the impeding concerns. Combustion rates limited by kinetics rather than transport and essentially no flame propagation lead to shorter charge consumption time and in turn higher operational engine speeds are allowed. Auto ignition implies that the charge is consumed uniformly, consequently minimizing quenching effects. Both these traits essentially bring combustion closer to a constant volume procedure pushing the thermodynamic process closer to an ideal

Otto cycle. No external ignition system is required absolving both the timing issues at the high speeds and the need for micro-scale spark inducers that fit the mechanism. The ability to sustain combustion with very lean mixtures reduces fuel mass losses via leakage between the piston and cylinder, adversely enhancing fuel consumption efficiency[6,13].

Because of the high surface to volume ratio of miniature IC engine, the friction between the sealing rings and the cylinder walls become a major factor, and in some cases are in the order of the engine power. For this reason, no rings are practically installed in these engines. Consequently, high precision manufacturing is needed in order to minimize the piston-cylinder walls gap. Practically $10-20\mu m$ is feasible. In these engines charge leakage is therefore unavoidable, and thus high engine speed is needed to lessen the losses incurred. However, when the piston velocity is faster, hydrodynamic friction losses between the piston and the cylinder-walls become more and more important. Sher et al.[17] proposed that the lower possible miniaturization limit of the HCCI engines is predominantly determined by the charge leakage through the cylinder-piston gap. A comprehensive mathematical model has been developed to characterize the relevant processes inside the cylinder (including the cylinder-piston gap) and then solve numerically to yield the inter-relationships between the charge leakage through the cylinder-piston gap, the engine geometrical dimensions and the engine speed. They found that the minimal practically possible size of a miniature HCCI engine depends on two major parameters; the engine speed and the gap width between the piston and the cylinder walls. It was postulated that since no rings are involved in miniature engines, the higher the engine speed and the smaller the cylinder -piston gap width are, the higher the engine efficiency is. However, in practical terms, the engine speed is limited by the strength of material from which the mechanical mechanism is made of (connecting rod, bearings, piston pin and crank). The gap width is practically limited by the fabrication method, thermal expansion coefficient and mechanical unsymmetrical stresses.

In the present work we propose an approximate analytical solution to this problem. An approximated analytical solution is proposed to yield the lower possible limits of scaling-down HCCI cycle engines. The present work presents a simple algebraic equation that shows these inter-relationships, thus constituting analytical expressions to postulate the lower possible miniaturization limits of IC engines.

Theoretical Consideration

In this section a phenomenological approximate model is developed to consider the relevant processes inside the cylinder of a Homogeneous-Charge Compression-Ignition (HCCI) engine. Figure 1 shows a schematic sketch of a simple SI crankcase-scavenged two-stroke engine[18]. Special attention is drawn to the inter-relationships between the charge leakage through the cylinder-piston gap, the engine geometrical dimensions and the engine speed. The cycle has been simplified and idealized in order to constitute the lowest possible scaling-down limit of an HCCI engine.

Basic assumptions

1. HCCI two-stroke engine with no piston rings.

2. In order to find the lowest scaling down law, except for the gas exchange process and possible charge leakage, an ideal Otto cycle is considered: combustion occurs in a constant volume process, and compression and expansions strokes are adiabatic.
3. The fresh charge and the burnt gas, each, behaves as an ideal gas with c_p=*const.*
4. The scavenging is a perfect mixing process.
5. The flow through the gap is isothermal, compressible, one-dimensional, and quasi-static.
6. The leakage reduces the effective compression process. The minimum effective compression process is determined by the lowest allowed compression temperature which must be higher than the fuel ignition temperature, thus, $T_{compression} > T_{ig}$.

Based on a previous study[13], it is postulated that the engine scaling-down limit is determined predominantly by the amount of charge induced and the amount of leakage through the gap between the piston and the cylinder. It is thus expected that at low engine speeds the amount of charge leakage has an important effect on the engine power, while at higher engine speeds the amount of leakage per cycle reduces, and the cylinder charging efficiency and engine friction losses dominate the engine performance.

Estimation of the cylinder charging efficiency and engine power

The scavenging process is assumed to be a perfect mixing process[18], and thus the charging efficiency of the cylinder is:

$$\eta_{ch} = \frac{m_{cylinder}}{m_{cylinder\_\max}} = 1 - \exp\left(- \frac{\Delta t_{charging}}{\tau} \right) \tag{1}$$

Where τ is the time period during which the scavenging ports open. A characteristic crank angle interval of 120CA[18] is considered, and thus the time interval during charging is $\Delta t_{charging} = \frac{1}{3}\frac{2L}{\bar{u}_{piston}}$. The maximum cylinder charge is by definition: $m_{cylinder\_\max} = \frac{\pi D^2 L}{4}\rho_0$. Following Heywood and Sher[18], the typical effective scavenging port area is half of the cylinder cross section area. A fair estimation of the characteristic time required to fill the cylinder may be obtained when the charging velocity is assumed to be close to sonic. Thus, $\tau = \frac{1}{a_0}\frac{Maximun\ trapped\ volume}{Effective\ Scavenging\ Port\ Area} = \frac{2L}{a_0}$, and eq. (1) becomes:

$$\eta_{ch} = 1 - \exp\left(- \frac{1}{3}\frac{a_0}{\bar{u}_{piston}} \right) \tag{2}$$

Where a_0 is the sonic velocity at ambient conditions.

The mass loss during compression may be estimated from $m_{leak} = \pi D \delta \rho_c a_c \frac{L}{\bar{u}_{piston}}$ where $\pi D \delta$ is the cross-section area of the leakage flow, ρ_c the charge density, a_c the sonic velocity at cylinder conditions, and $\frac{L}{\bar{u}_{piston}}$ is the characteristic available time. The mass loss fraction is thus:

$$\alpha[mass\ loss\ \ fraction] = \frac{\pi D\delta\rho_c a_c \frac{L}{\bar{u}_{piston}}}{\frac{\pi D^2}{4} L\rho_0\eta_{ch}} = \frac{4a}{\eta_{ch}\bar{u}_{piston}}\frac{\delta}{D}\frac{\rho_c}{\rho_0} \tag{3}$$

Where $\frac{\rho_c}{\rho_0} = \left(\frac{T_{ig}}{T_0}\right)^{\frac{1}{k-1}}$. The engine indicated power may be estimated from the difference between the ideal indicated power and the power needed to perform charge leakage, as follows:

$$ip = \eta_{th} \cdot \eta_{Ch} \cdot (1-\alpha)\frac{\pi D^2\bar{u}_{piston}}{8}\rho_0 h - imep \cdot \pi D\delta a \tag{4}$$

Where $imep = \frac{ip}{\frac{\pi D^2 L\bar{u}_{piston}}{8}}$ and $h = \frac{FA}{1+FA}h_c$. Thus:

$$ip = \frac{\eta_{th} \cdot \eta_{Ch} \cdot (1-\alpha)\frac{\pi D^2}{8}\rho_0 h\bar{u}_{piston}}{1+\frac{8\delta a_c}{D\bar{u}_{piston}}} \tag{5}$$

Estimation of the engine friction losses

Following Heywood and Sher[18], the total friction mean effective pressure of a two-stroke engine may be estimated from a simple quadratic equation of the form of:

$$tfmep[kPa] = a + b\left(\frac{N[rev/\min]}{1000}\right) + c\left(\frac{N[rev/\min]}{1000}\right)^2 \tag{6}$$

Where a~190kPa, b~-20kPa*min, c~3.6kPa*min^2. The power losses due to the friction between the cylinder skirt, the rings and the cylinder wall are represented mainly by the 3rd term of eq. (6), and therefore this parameter is to be calibrated to fit a no rings engine. The engine brake power, by definition, is the difference between *ip* (eq. 5) and *fp* (eq. 6) thus, $bp = ip - fp$.

Model Calibration

The mathematical model has been calibrated to fit experimental results of two small bore engines. The first engine, TD .020, is one of the smallest 2-stroke engines ever produced (the only smaller one known to the authors is the TD .010 produced in 1997. Its bore=0.602cm, stroke=0.574 and swept volume=0.1634cc. No formal data is available for this engine). The 2nd engine, TD 0.15 is another small bore engine for which relevant data are available. The specifications of the two engines are tabulated in the following table:

Table 1: Engine specifications

Engine type	COX 2-stroke	COX 2-stroke
Engine model	TEE-DEE .02	TEE-DEE .15
Scavenging system	Schnurle	Schnurle
Charging	Naturally aspirated	Naturally aspirated
Bore [cm]	0.762	1.486
Stroke [cm]	0.716	1.488
Swept volume [cc]	0.3266	2.45
Peak power output [W]	41.0 at 23,200rev/min	260.0 at 17,000rev/min

Figure 2 shows how the 3rd term friction coefficient - *c* in eq. (6), affects the prediction of the specific power curve of these particular engines. Also shown are some available experimental results[19]. The effect of the charging deterioration is clearly seen for the case of a hypothetical engine without friction (c=0), where the specific power deviate from a linear dependence on the engine speed (in fact, the 1st and 2nd terms of eq. (6) are not zero, but the 1st is a constant and doesn't depend on the engine speed, while the 2nd contributes a negative value to the friction losses). Beside of the breathing constrains, the friction losses play a major role in shaping the performance curves of any engine, and it seems that c=2.2 fits best the experimental results of the two COX engines.

Results and Discussion

In order to achieve good scavenging in small two-stroke engines, the cylinder length to the cylinder bore ratio should be around a unity[18]. In the following discussion (except for the two specific engines for which exact dimensions are provided) we assume L~D, and thus the engine characteristic dimension is the engine bore. In HCCI engines, ignition occurs when the temperature of the cylinder charge reaches the ignition temperature. For an adiabatic cylinder, the in-cylinder temperature prior to ignition depends on the compression ratio and the charge fraction loss due to leakage. The charge fraction loss due to leakage depends on the engine dimensions (bore and ports heights), gap width and engine speed. It follows that for a particular geometry, the engine speed must exceed a certain value such that the time allowed for leakage is short enough to obtain the required effective compression, as illustrated in Fig. 3a for a cylinder bore of 3cm. As expected, a wider gap width results in a higher required compression ignition to ensue ignition at a given engine speed. The required compression ratio increases at low engine speeds and approaches a constant value at the higher engine speeds. For wider gaps the requirement becomes higher in particular at low engine speeds. The corresponding specific power of the engine, i.e., power to volume ratio, increases as the gap width decreases (Fig. 3b). At low engine speeds, it increases nearly linearly with the engine speed, while at higher engine speeds the specific power reaches a maximal value and then drops to zero. The latter is attributed to the deterioration of the charging efficiency and the increasing friction power losses as the engine speed increases.

The effect of the engine size on the required compression ratio and corresponding specific power for a gap width of 20μm, is shown in Figs. 4a and 4b, respectively. As compared to the larger engine with the same gap width, in the smaller engines the charge fraction loss is higher and the minimum required piston speed increases. This may explain the experienced starting difficulties of small engines w/o glow plugs, in which high speed starters are needed. Figure 4 also suggests that the engine speed at which the engine develops its maximum power depends on the engine bore. Closer examination revealed that it doesn't depend noticeably on the gap size though its maximal power does significantly as shown soon after. Figure 5 shows the relationship between the speed at which the engine develops its maximum power and the engine bore for miniature engines.

The effects of the gap width between the piston and the cylinder wall on the minimum required compression ratio and specific power is demonstrated in Figs. 6a and 6b for a miniature engine as described in table 1. As compared to the hypothetical case for which no gap is involved, the maximum engine power drops dramatically from above 200W/cc to below 50W/cc when the gap increases to 20μm, which may explain the traditional low overall efficiencies of energy conversion (to distinguish from the thermal cycle efficiencies) of miniature engines. The required compression ratio for maximum power increases from around 6 to 16, while the speed range of engine operation becomes noticeably narrower.

Figure 3 shows how the specific power and maximum engine speed depend on the gap width for a cylinder bore of 3cm. It is noted that while the maximum engine speed is not realistic, the engine speed at which maximum power is obtained is an important and practical parameter for such small size engines. Figure 3 also suggests that the gap width is a crucial parameter in scaling down engines; a gap width above 20μm casts a severe limitation for the engine size. It is anticipated that for such high engine speeds a high volatile, low ignition temperature and low ignition delay fuels are essentially required. Figure 7 shows how the ignition temperature of the fuel affects the required minimum compression ratio and minimum engine speed, for a small bore engine of 0.762cm with a gap width of 10μm. The value of 0.762cm corresponds to a 0.3266cc engine which represents one of the smallest available 2-stroke engines in the market. The typical fuel for a miniature two-stroke engine is a 30% nitro-methane fuel with an ignition temperature of around 600K. The ignition temperature of the fuel is clearly an important issue for the engine designer.

Figure 8 shows some important results that can be drawn from the present study; it shows the maximum specific power of a miniature engine vs. the gap width and its cylinder bore. The corresponding engine speeds are shown in Fig. 5. The following conclusions may be postulated: 1) the engine power decreases as the gap width increases. 2) As the engine size decreases the power output is less sensitive to the gap width. 3) For a gap width of 10μm (which is typical to the present surface texture), the minimum possible engine size is 0.4cm bore (corresponds to 0.05cc). 4) For a more realistic gap width of 20μm (which reflects thermal distortions in well-designed engines), the minimum possible engine size is 0.7cm bore (corresponds to 0.27cc). The COX TD .010 is the smallest engine ever produced commercially had a bore=0.602cm and a stroke=0.574 (corresponds to a swept volume=0.1634cc). Fig. 8 suggests a maximum gap width=16mμ for this engine. 5) For smaller engines of 0.3 or 0.2cm bore, the engine should be manufactured such that the gap width at operation conditions must be as small as a few microns, while engines' speeds at maximum power are expected to be higher than 30,000rev/min. Even though such a small gap can hardly be achieved today in practice without sealing rings, it is still a challenging and interesting task for the future.

The lower possible limits of scaling-down HCCI cycle engines can be fairly estimated for high engine speeds, which are typical for miniature engines with the following additional assumptions. For this purpose we assume that the 3rd term of eq. (6), which represents the power losses due to the friction between the cylinder skirt, the rings and the cylinder wall, becomes dominant. Considering that $fp = tfmep\frac{\pi D^2 L}{4}\frac{N}{60}$, and $N = 60\frac{\bar{u}_{piston}}{2L}$, eq. (6) yields:

$$fp = c\frac{\pi 10^{-6} 60^2}{32}\left(\frac{D}{L}\right)^2 \bar{u}^3_{piston} \tag{7}$$

Furthermore, figs. 3 and 4 show that for the examined cases the maximum power of the engine is obtained at about half of its maximum possible speed. Therefore, the maximum engine speed, which is evidently achieved when $ip = fp$, may be derived from the following equation:

$$\begin{gathered}\eta_{th}\eta_{Ch}(1-\alpha)\frac{\pi D^2 L}{4}\rho_0\frac{FA}{1+FA}h_c\frac{N}{60} = c\frac{\pi 10^{-6} 60^2}{4\cdot 8}\left(\frac{D}{L}\right)^2\left(\bar{u}^3_{piston}\right)_{\max} \\ \text{or :} \\ \left(\bar{u}^2_{piston}\right)_{\max} = \eta_{th}\left\{1-\exp\left[-\frac{1}{3}\frac{a_0}{\left(\bar{u}_{piston}\right)_{\max}}\right]\right\}\left[1-\frac{\delta}{D}\frac{\rho_c}{\rho_0}\frac{4a}{\left(\bar{u}_{piston}\right)_{\max}}\right]\frac{L^2\rho_0}{30^2 10^{-6} c}\frac{FA}{1+FA}h_c\end{gathered} \tag{8}$$

Equation (8) is an approximate implicit equation that relates the maximum possible engine speed to the cylinder dimensions and gap width. Equation (8) constitutes the lower possible miniaturization limits of IC engines.

Conclusions

In the present work a phenomenological model has been developed to consider the relevant processes inside the cylinder of a Homogeneous-Charge Compression-Ignition (HCCI) engine. The inter-relationships between the pertinent parameters (the charge leakage through the cylinder-piston gap, the engine geometrical dimensions and the engine speed), has been established. An approximated analytical solution is proposed to yield the lower possible limits of scaling-down HCCI cycle engines. The present work presents simple algebraic equations that shows the inter-relationships between the pertinent parameters, and constitutes the lower possible miniaturization limits of IC engines. We show that the minimal practically possible size of a miniature HCCI engine depends on two major parameters; the engine speed and the gap width between the piston and the cylinder walls; the higher the engine speed and the smaller the cylinder piston gap width are, the higher the engine energy conversion efficiency is. However, in practical terms, the engine speed is limited by the strength of material from which the mechanical mechanism is made of (connecting rod, bearings, piston pin and crank). The gap width is practically limited by the fabrication method, thermal expansion coefficient and mechanical asymmetrical stresses. Current technology allows an engine speed of up to around 40,000 to 60,000rev/min, while a gap width of 6-8μm is hardly feasible. Under these conditions, we found that the minimum allowed engine size is between 0.3 and 0.4cc.

References

[1] Cadou, C. and Menon, S., "Scaling of Losses in Small IC Aero Engines with Engine Size", *42nd AIAA Aerospace Sciences Meeting and Exhibit, Reno*, AIAA-2004-0690, Reno NV, Jan, 2004.

[2] Hua, J., Wu, M. and Kumar, K., "Numerical simulation of the combustion of hydrogen-air mixture in micro-scaled chambers. Part I: Fundamental study", *Chemical Engineering Science*, Vol. 60, 2005, pp. 3497-3506.

[3] Lee C.H., Jiang K.C., Jin P. and Prewett P.D., "Design and fabrication of a micro Wankel engine using MEMS technology", *Microelectronic Engineering*, Vol. 73, 2004, pp. 529-534.

[4] Lee D.H. and Kwon S., "Heat Transfer and quenching analysis of combustion in a micro combustion vessel", *Journal of Micromechanics and Microengineering,* Vol. 12, 2002, pp. 670-676.

[5] Aichlmayr, H.T., Kittelson, D.B. and Zachariah, M.R., "Miniature free-piston homogenous charge compression ignition engine-compressor concept-Part I: performance estimation and design considerations unique to small dimensions", *Chemical Engineering Science*, Vol. 57, 2002, pp. 4161-4171.

[6] Aichlmayr, H.T., Kittelson, D.B. and Zachariah, M.R., "Micro-Homogeneous Charge Compression Ignition (HCCI) Combustion: Investigations Employing Kinetic Modeling and Experiments", *Combustion Institute Spring Technical Meeting*, La Jolla, CA, March 2002.

[7] Aceves, S.M., Flowers, D.L. and Dibble, R., "HCCI Combustion: Analysis and Experiments", U.S. Government Works, 2001-01-2077, 2001.

[8] Cadou, C., Menon, S. and Moulton, N., "Performance Measurement and Scaling in Small Internal Combustion Engines", 41st AIAA Aerospace Sciences Meeting and Exhibit, Reno, AIAA 2003-0671, Reno NV, Jan, 2003.

[9] Menon S., Moulton N. and Cadou C., "Performance Measurements of Small Internal-Combustion Engines", *Journal of Propulsion and Power*, September, 2005.

[10] Hua, J., Wu, M. and Kumar, K., "Numerical simulation of the combustion of hydrogen-air mixture in micro-scaled chambers. Part II: CFD analysis for a micro-combustor", *Chemical Engineering Science*, Vol. 60, 2005, pp. 3507-3515.

[11] Dahoe, A.E. and de Goey, L.P.H., "On the determination of the laminar burning velocity from closed vessel gas explosions", *Journal of Loss Prevention in the Process Industries*, Vol. 16, 2003, pp. 457-478.

[12] Ilbas M., Crayford A.P., Yilmaz I., Bowen P.J. and Syred N., "Laminar Burning Velocities of Hydrogen-Air and Hydrogen-Methane-Air Mixtures: An Experimental Study", *International Journal of Hydrogen Energy*, 2006.

[13] Sher, E. and Levinzon, D., "Scaling-Down of miniature Internal Combustion Engines: Limitations and Challenges", *Int' J. of Heat Transfer Engineering*, Vol. 26(8), 2005, pp.1-4.

[14] Onishi, S, Jo, SH., Shoda, K., Jo, PD. And Kato, S., "Active Thermo-Atmosphere Combustion (ATAC) – A New Combustion Process for Internal Combustion Engines", SAE Paper 790501, 1979.

[15] Spadassini C.M., Zhang X., Cadou C.P., Miki N. and Waitz I.A., "Preliminary development of a hydrocarbon-fueled catalytic micro-combustor", *Sensors and Actuators*, Vol. 103, 2003, pp. 219-224.

[16] Aichlmayr, H.T., Kittelson, D.B. and Zachariah, M.R., "Miniature free-piston homogenous charge compression ignition engine-compressor concept-Part II: modeling HCCI combustion in small scales with detailed homogeneous gas phase chemical kinetics", *Chemical Engineering Science*, Vol. 57, 2002, pp. 4173-4186.

[17] Sher, I., Levinzon-Sher, D. and Sher, E., "Miniaturization Limitations of HCCI Internal Combustion Engines", *Applied Thermal Eng.*, Vol. 29, 2009, pp. 400-411.

[18] Heywood, J.B., and Sher, E., *The Two-Stroke Cycle Engine - Its development, design and characteristics*, Joint pub. of SAE and Taylor and Francis, 1999.

[19] http://www.mh-aerotools.de/airfoils/cox_teedees.htm

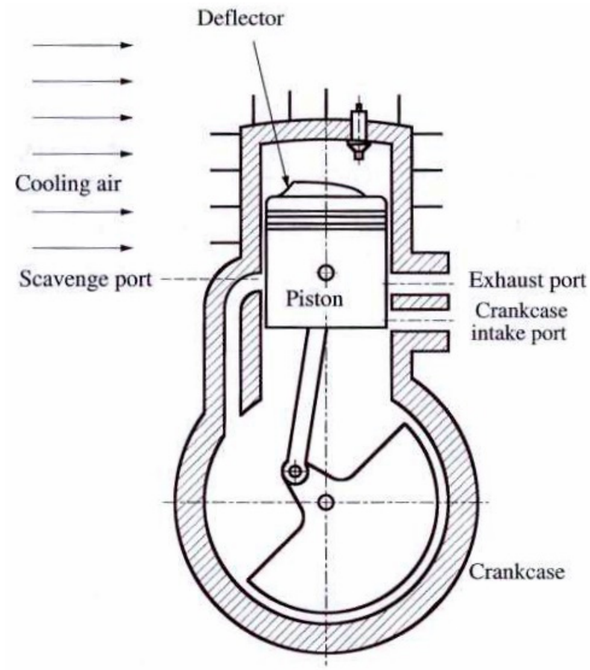

Figure 1: A schematic sketch of a simple SI crankcase-scavenged (Schnurle type) two-stroke engine[18].

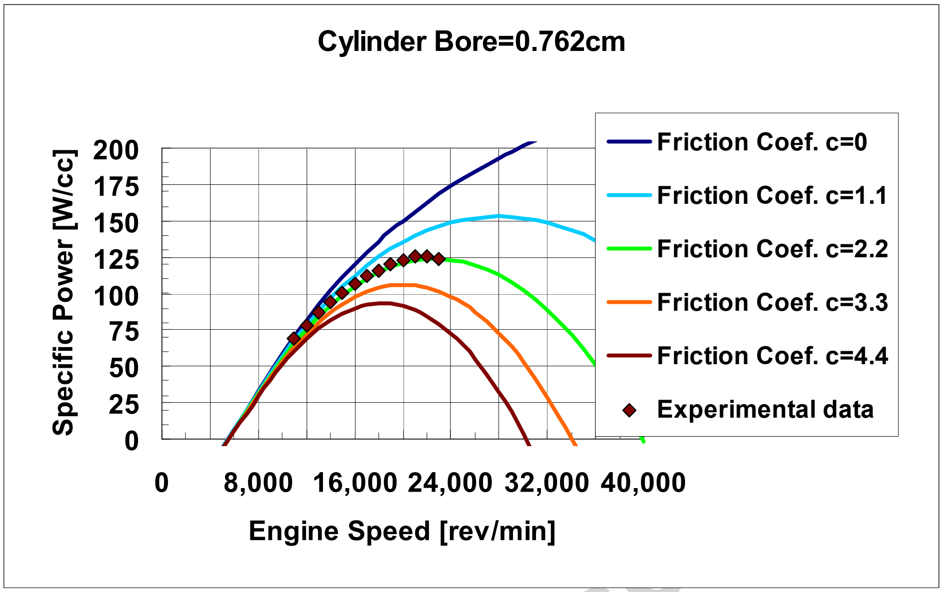

Figure 2a: The effect of the friction coefficient *c* in eq. (6) on the specific brake power of the COX TD.02 engine. The engine specifications are tabulated in table 1.

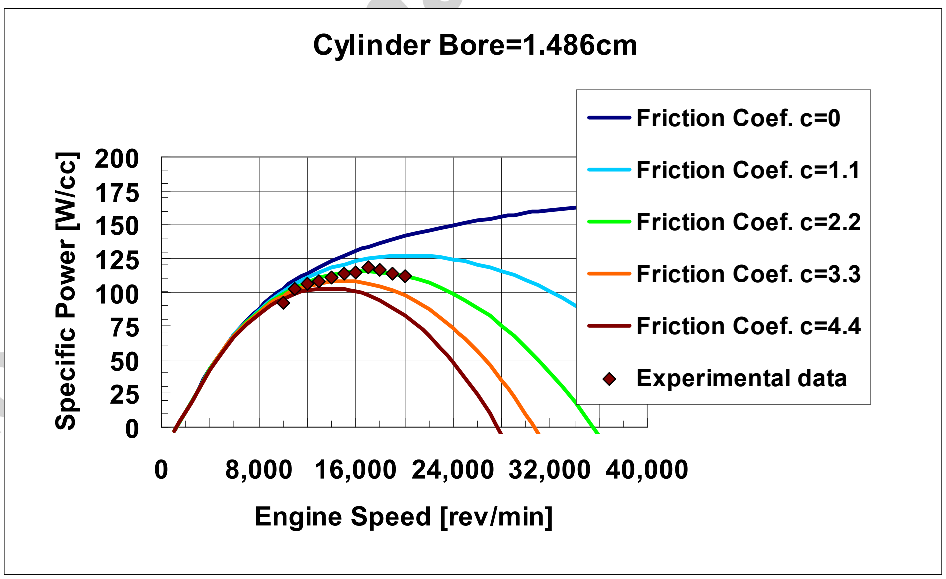

Figure 2b: The effect of the friction coefficient *c* in eq. (6) on the specific brake power of the COX TD.15 engine. The engine specifications are tabulated in table 1.

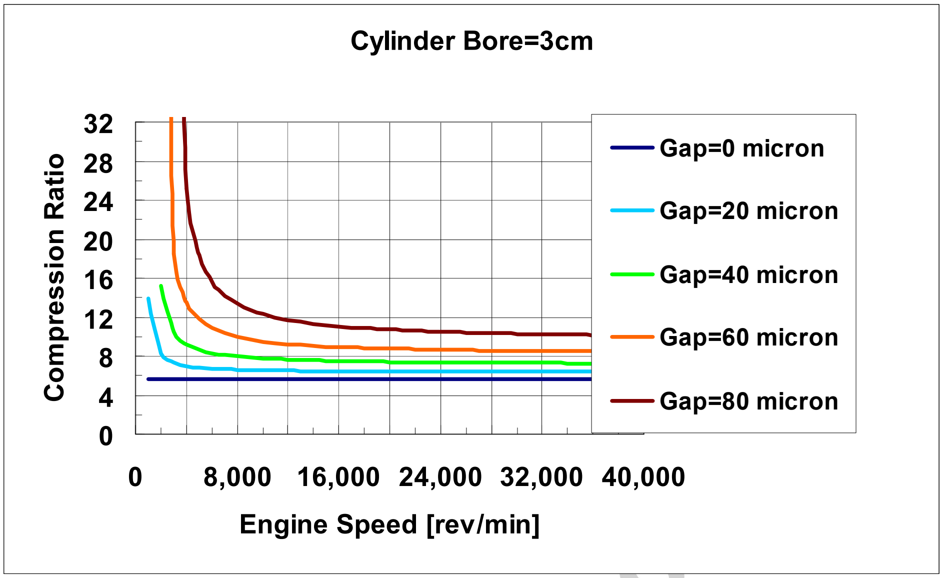

Figure 3a: Minimum compression ratio that is required to ensure ignition vs. engine speed and gap width for a cylinder bore of 3cm.

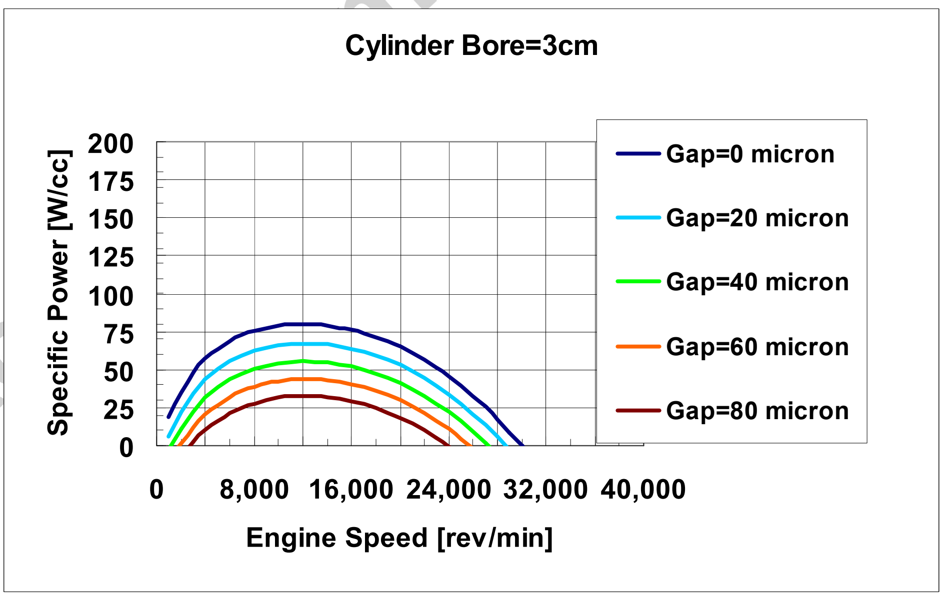

Figure 3b: The corresponding specific brake engine power.

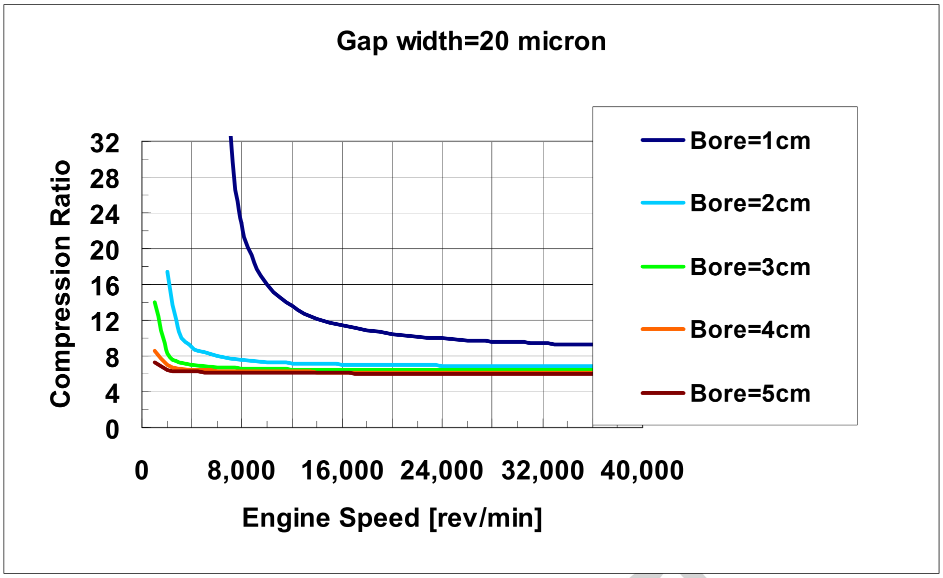

Figure 4a: Minimum compression ratio that is required to ensure ignition vs. engine speed and engine bore for a gap width of 20μm.

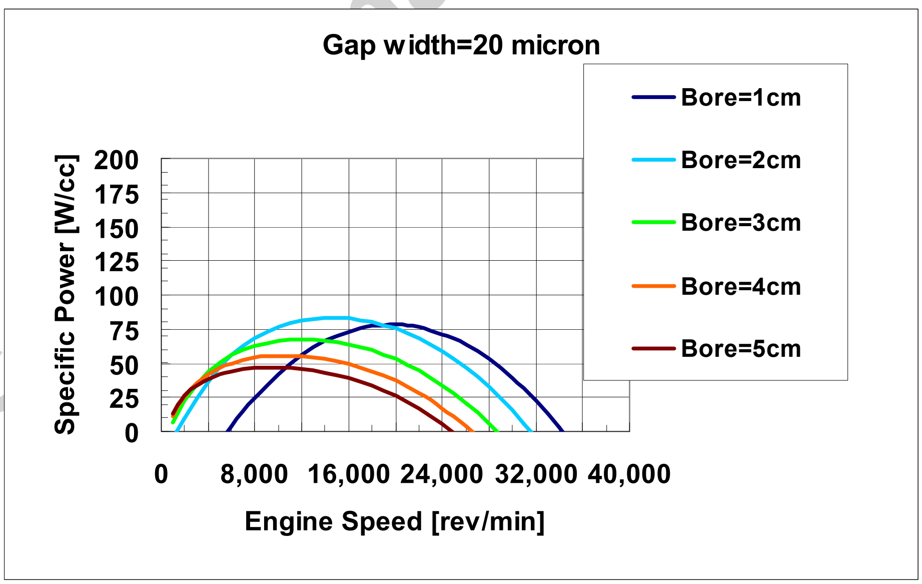

Figure 4b: The corresponding specific brake engine power.

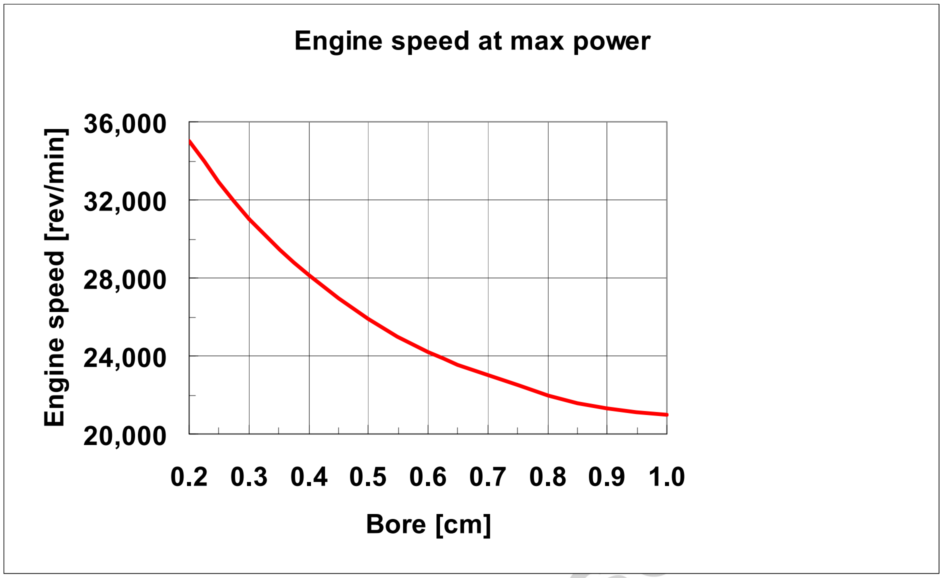

Figure 5: Engine speed at which the engine develops its maximum power vs. its bore. Closer examination revealed that it doesn't depend noticeably on the gap size

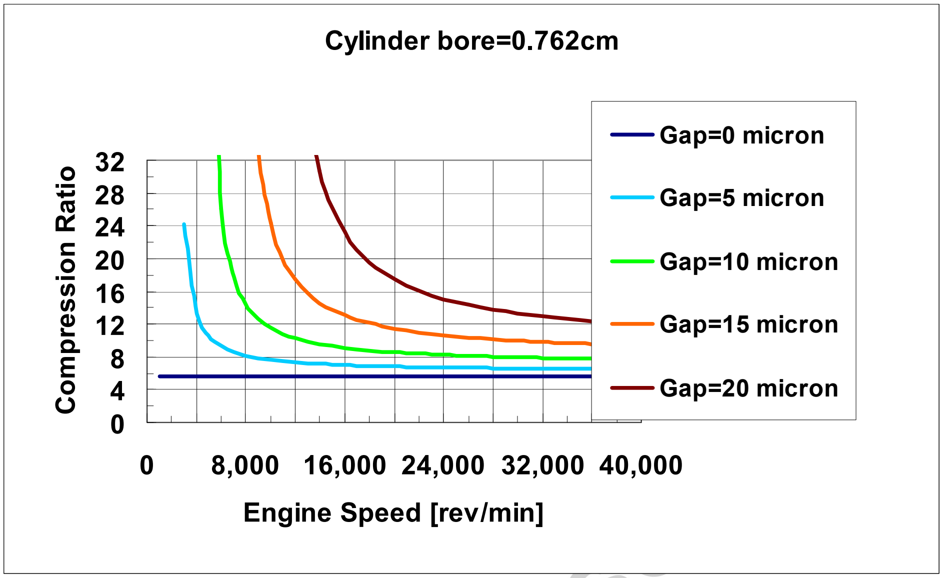

Figure 6a: Minimum compression ratio that is required to ensure ignition vs. engine speed and gap width for a small engine (bore=0.762cm).

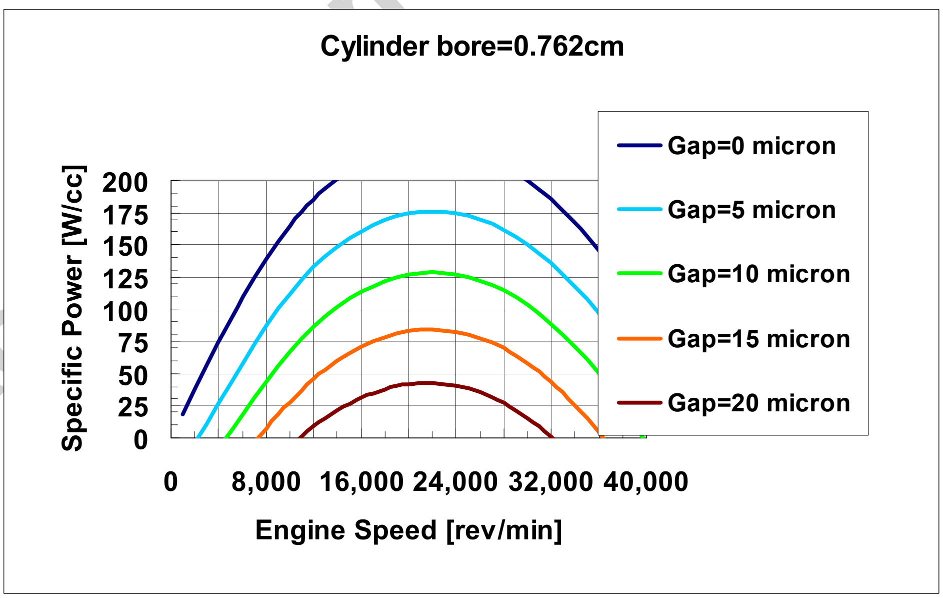

Figure 6b: The corresponding specific brake engine power.

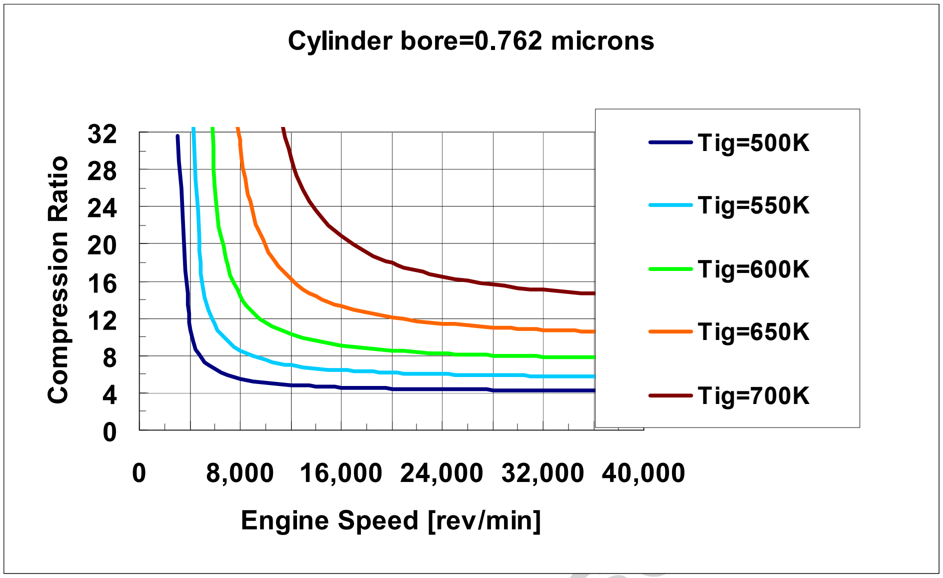

Figure 7: The effect of the fuel ignition temperature on the minimum compression ratio and minimum engine speed for a small size cylinder of 0.762cm and gap width of 5μm.

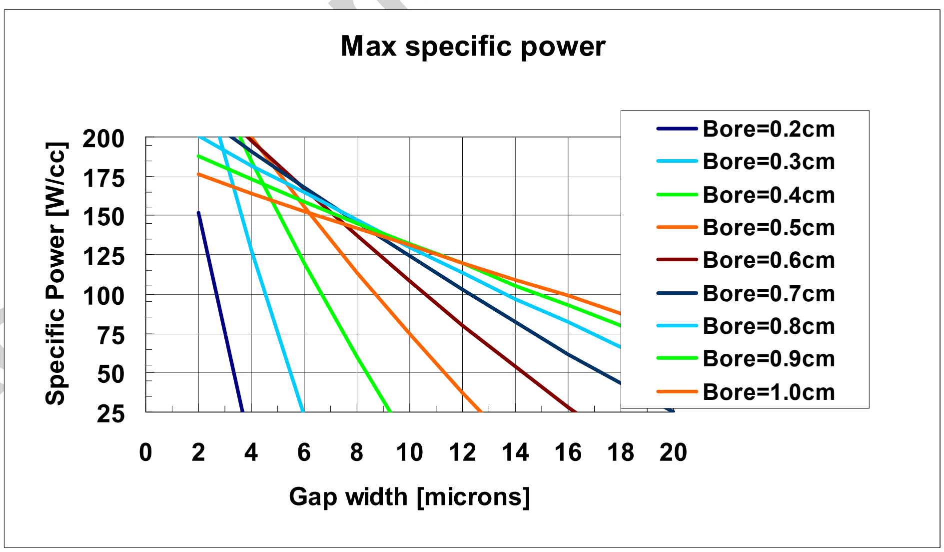

Figure 8: Maximum specific brake power vs. gap width and cylinder bore. The corresponding engine speeds are shown in Fig. 5.