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A Study on the Effect of Squish Area on Engine Performance of Single Cylinder Natural Gas Converted Engine

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Abstract

Today, the prices of fossil fuels such as gasoline and diesel are skyrocketing, oil depletion and air pollution are major challenges for us and the auto industry in particular. Natural gas has known as a potential alternative fuel for internal combustion engines because of its advantages such as the octane number, which is higher than that of gasoline, the low heat value which is higher in comparison with gasoline and diesel and the safety in use. This paper presents a study on the influence of squish area on engine performance of single cylinder natural gas converted engine. The obtained results indicated that the increase of compression ratio only augmented the risk of knocking for single cylinder natural gas converted engine. Conversely, the modification of bowl-in-piston is directly varied squish area, thus the turbulent kinetic energy of the gas flows at the end of the compression stroke increased in comparison with the flat head piston of the original engine.

Keywords: Natural gas, piston geometry, engine performance, converted engine.

1. Introduction

In recent decades, the economic growth of the world has led to the rapid increase of internal combustion engines [1]. Rising concerns about emissions have put great strain on the automotive industry. As a result, the industry is looking for next-generation engines and advanced combustion technology with extremely low emissions and high efficiency [2]. To achieve this, more understanding of combustion and mixture formation inside the cylinder is needed [3]. The research direction, using Compressed Natural Gas (CNG) as fuel for internal combustion engines has solved several problems such as: saving fossil fuels to ensure energy security, limiting emissions of greenhouse gases, protecting the environment, production, traffic, and daily life [4]. The main component of natural gas is methane (CH4) accounting for 85-96%, the rest is a small amount of ethane (C2H6), propane (C3H8), butane (C4H10), and a small number of other gases [5]. The emissions in combustion process products such as CO, particulate matter (PM), and NOx will be lower because of cleaner combustion and the combustions do not produce CH4 emissions, as CH4 is the main component [4].

In Viet Nam, the use of natural gas as fuel for internal combustion engines has gradually expanded and developed. The solution to convert a traditional diesel engine into a forced-ignition natural gas engine on the one hand allows taking advantage of the diesel engine's low speed and high compression ratio to improve engine performance with new fuel, on the other hand, solving the problem of production costs of new CNG engines. Due to natural gas existing in the form of gas, natural gas will be easier to mix with the air than liquid fuel (gasoline and diesel), so the amount of fuel loaded into the engine cylinder will burn more easily [7]. In addition, during operation, it does not consume liquid fuel to inject primer [8]. This helps to improve economic efficiency when using natural gas engines. However, because the fuel characteristics of CNG are different from the fuel form of traditional Diesel. Therefore, exploiting and using optimally the performance of the post-conversion engine is an extremely important issue. The aim of this study is to analyze the effect of bowl-in-piston on the performance of a converted Diesel engine using CNG fuel forced combustion - SING engine. From the above issues, it shows that this research is necessary for today's actual situation.

2. Theoretical Framework

A theoretical squish velocity can be calculated from the instantaneous displacement of gas across the inner edge of the squish region (across the dash lines in the drawings in Fig. 1). The original diesel engine's cylinder head and piston top are both flat. Ignoring the effects of gas dynamics (non-uniform pressure), frictions, leakage past the piston rings, and heat transfer, the squish velocity's expression is

$$\frac{v_{sq}}{S_p} = \frac{D_b}{4z}\left[\left(\frac{B}{D_b}\right)^2 - 1\right]\frac{V_b}{A_c z + V_b} \tag{1}$$

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$$\frac{\partial k}{\partial t}+\overline{u_j}\frac{\partial k}{\partial x_i}=-\frac{1}{\rho_0}\frac{\partial\overline{u'_i p'}}{\partial x_i}-\frac{1}{2}\frac{\partial\overline{u'_j u'_j u'_i}}{\partial x_i}+v\frac{\partial^2 k}{\partial x_j^2}-\overline{u'_i u'_j}\frac{\partial\overline{u_i}}{\partial x_j}-v\overline{\frac{\partial u'_i}{\partial x_j}\frac{\partial u'_i}{\partial x_j}}-\frac{g}{\rho_0}\overline{\rho' u'_i}\delta_{i3} \quad (2)$$

where V_b is the volume of the piston bowl (m^3), A_c is the cross-sectional area of the cylinder (m^2), S_p is the instantaneous piston speed (v^2/m), z is the distance between the piston crown top and the cylinder head (m), l is connecting rod length (m), a: the crank radius (m), s is the distance between the crank axis and the piston pin axis, c: the clearance height, D_b: the diameter of the bowl, H_b: the depth of the bowl.

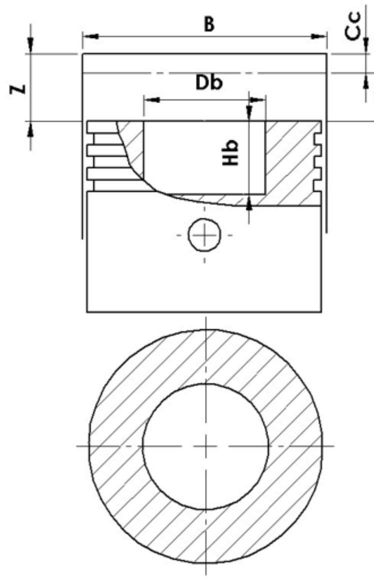

Fig. 1. Schematic of the bowl-in-piston chamber and squish area.

Turbulent Kinetic Energy (TKE) is the average kinetic energy per unit mass with circular swirls of turbulent flow. This vortex tends to run into large spaces and has lower pressure. For the refrigerant flow inside the engine cylinder with stable viscosity, the turbulent kinetic energy (TKE) equation of the refrigerant flow inside the engine cylinder of the mixture of air and natural gas is written as equation (2):

where: $\frac{\partial k}{\partial t}$ is local derivative; $\overline{u_j}\frac{\partial k}{\partial x_i}$ is advection; $\frac{1}{\rho_0}\frac{\partial\overline{u'_i p'}}{\partial x_i}$ is pressure diffusion; $\frac{1}{2}\frac{\partial\overline{u'_j u'_j u'_i}}{\partial x_i}$ is turbulent transport (T); $v\frac{\partial^2 k}{\partial x j_j^2}$ is molecular viscous transport.; $-\overline{u'_i u'_j}\frac{\partial\overline{u_i}}{\partial x_j}$ is production (P); $v\overline{\frac{\partial u'_i}{\partial x_j}\frac{\partial u'_i}{\partial x_j}}$ is dissipation $(\mathcal{E}_k)$; $\frac{g}{\rho_0}\overline{\rho' u'_i}\delta_{i3}$ is buoyancy flux (b).

An important parameter that also needs to be considered and evaluated through measurement parameters to evaluate the quality of combustion is Mass Fraction Burned (MFB). The value of MFB is calculated based on the ratio between the accumulated heat of the fuel released from the combustion process to the total theoretical heat of the fuel injected into the engine cylinder. The burned fuel mass factor is a function that varies with the crankshaft rotation angle, the formula is as follows:

$$MFB=\frac{\int_{\theta_{soc}}^{\theta}\left\{\frac{\delta Q_{gen}}{d\theta}\right\}d\theta}{m_{f,total}\times\eta_{comb}\times Q_{LHV}} \quad (3)$$

Where: MFB is Mass Fraction Burn; θ is the crankshaft rotation angle (radial); Q_{gen} is the total theoretical heat of the fuel injected (kJ); $m_{f,total}$ is the total intake fuel mass (g/s); η_{comb} is thermal efficiency; Q_{LHV} is the low heating value, (kJ/kg).

Heat release rate (HRR) is the rate at which heat is released during the combustion of fuel in an engine cylinder. Based on the HRR value, it is possible to evaluate the characteristics of the fuel combustion process inside the engine cylinder and diagnose the composition of the exhaust gases formed. The heat release rate is calculated based on the 1st law of thermodynamics with the non-dimensional and mixed kinematics model in a single-zone cylinder, from the pressure parameter in the cylinder measured at 100 cycles, the HRR can be calculated according to the following general formula:

$$\frac{dQ_c}{d\theta}=P\left(\frac{\gamma}{\gamma-1}\right)\frac{dV}{d\theta}+V\left(\frac{1}{\gamma-1}\right)\frac{dP}{d\theta}+\frac{dQ_h}{d\theta} \quad (4)$$

where: $\frac{dQ_c}{d\theta}$ is heat released from combustion process in engine cylinder.

$\frac{dQ_h}{d\theta}$ is heat transfer to wall of combustion chamber.

To prevent the auto-ignition phenomenon in spark-ignition engines, it is needed to determine the knocking limit by combining the maximum pressure value and the required octane number (ON). The required octane number is considered as the following formula

$$ON=100.\left(\frac{1}{A}\int_{t_{SOC}}^{t_{85\%MBF}}\left[\left(\frac{p}{p_{Ref}}\right)^n.exp\left(-\frac{B}{T_{UBZ}}\right)\right]dt\right)^{\frac{1}{a}} \quad (5)$$

3. Engine Simulating, Calibration and Controlling Model

3.1. Experimental Setup

Experimental setup is an important step to collect the parameters on the test bench, which will be used to calibrate the model. Research equipment and engine were arranged as shown in Fig. 2 and 3, including the following equipment: Ricardo single-cylinder research engine redesigned from a horizontal single-cylinder diesel engine with the parameters presented in Table 1. The CNG fuel supply system (Mass Flow Controller: MFC) and a port CNG injector, a Dynamometer was used to measure the engine's torque, in addition, there were the intake/exhaust system, the cooler system, the engine control unit, the data collector and others measuring systems.

Table 1. Basic parameters of QTC2015

Parameter	Symbols	Value
Bore, (mm)	D	103
Stroke, (mm)	S	115
Injector (-)	i	1
Number of Stroke (-)	T	4
Compression ratio	E	9-35

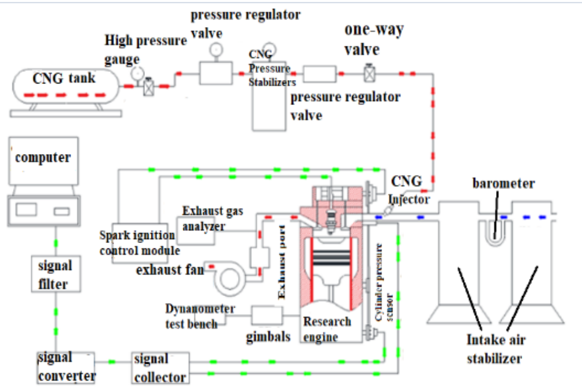

Fig. 2. Scheme of the experimental equipment setup


Fig. 3. Engine and experimental equipment

3.2. Simulating theoretical framework

The Fractal Combustion Model was selected as the research model for the mixed charge flow from the AVL Boost software's library. This was the suitable model for CI engines [9], the theoretical framework is summarized below.

Ignition timing was considered as the start of the combustion of simulation. The flame front formation was the parameter to calibrate the ignition delay (C_{ign}). The flame propagation speed was the parameter to calibrate the ignition delay ($r_{f,ref}$). The burned mass of fuel in a time unit was calculated as the formula:

$$\frac{dm_b}{dt} = \rho_u \left(\frac{L_1}{l_k}\left(\frac{\rho_{soc}}{\rho_{uz}}\right)^m\right)^{D_3-2} \times A_L \times S_L \qquad (6)$$

where: m is the calibration parameter of turbulence model; ρ_{soc} is the unburn density at the start of combustion; ρ_{uz} is the unburn density.

The small amount of burned mass at the start of wall combustion determined in-wall combustion process was $\left(\frac{m_b}{m}\right)_{tr}$, where the transition time t_{tr} has been determined when a small amount of mass was burned. The laminar burning speed $S_L = c_{lfs}\, S_{L,RG=0}\left(1 - m_{fRG}\right)^d$ has been determined at the start of wall combustion (d), allowed to adjust more S_L depending on residual gas mass coefficiency (m_{fRG}).

Fig. 4 presented the elements of the QTC 2015 engine simulated by AVL Boost software, each element of the simulation engine had the same parameters as the experimental engine.

Based on the QTC2015 engine structural parameters, CNG test fuel and AVL Boost software manual, the one-way model of the engine is shown on Fig. 4, and annotation of the elements in Table 2.

Table 2. Element name of the simulated motor

Element name	Symbols	Amount
Engine	E	1
Cylinder	C	1
Air filter	CL	1
Throttle	TH	1
Injector	I	1
Plenum	PL	1
Measuring points	MP	7
Restriction	R	3
System Boundary	SB	2
Pipes	→	8

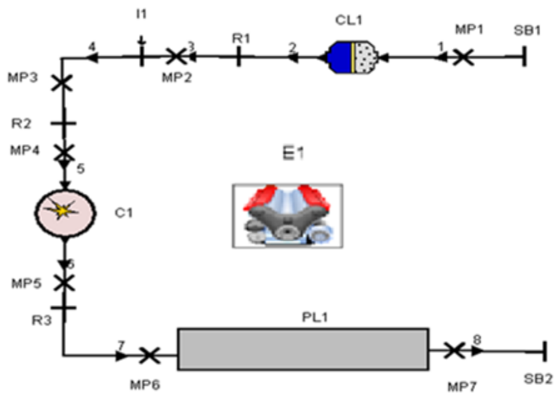

Fig. 4. The simulation engine QTC2015

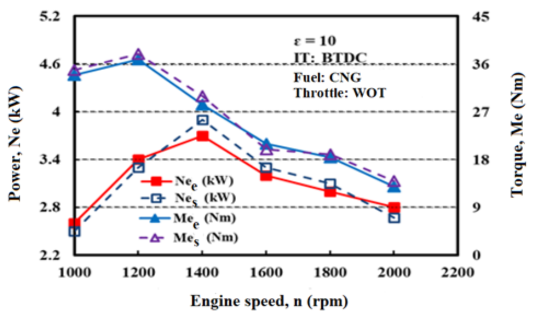

Fig. 5. The calibration results of the model

3.2. Model Calibration

Fig. 5 presented the results such as torque (M_e) and power (N_e) of the experimental and simulation engine, with the solid lines were the results of the real engine on the test bench. The dash lines represented the simulation model's results after recalibrating the model. However, the parameters of QTC2015 experimental engine such as cylinder bore, piston parameters, stroke, lengths, and diameters of intake and exhaust ports were used to input for the model.

The experimental condition of the test engine is wide-open throttle (WOT) so this element wasn't used in the model, spark angle was adjusted before top dead center (IT: BTDC) and compression ratio is ε = 10. Considering the whole experimental zone (n = 1000 - 2000 rpm), the maximum and minimum errors between the simulation results and experimental results were about 5% and 2%. However, at the speed n = 1800 rpm, the errors of both torque and power were approximately 2% and this speed was fixed to study the influences of the structuring parameters on combustion duration.

3.3. Controlling the Model

To consider the effect of the bowl in piston on the piston top on the SING engine's performance, the simulation study will be proceeded as follows: the port injection pressure is kept constantly with P_f = 1, the throttle is fully opened (Throttle: WOT) to reduce the losses.

The center of the bowl volume on the top of the piston top and the spark plug center coincides with the center line of the engine cylinder. Structuring parameters were varied: the bowl depth with H_b = 0 (Piston shape: Flat), H_b = 10 mm and H_b = 17 mm. Meanwhile, the bowl diameter was varied: D_b = 0 (Flat-peak piston), D_b = 60 mm and D_b = 66 mm. Engine speeds were varied: n = 1000-2200 rpm with a step n = 200. The compression ratio ε = 10-15 changed until the ON value > 130 then stopped. To study the effect of bowl in piston on the combustion and heating characteristics in the cylinder, the engine speed is n = 1800 rpm, λ is constant, ignition timing is chosen to achieve the maximum brake torque IT = MBT.

Table 3. Structuring parameters of study piston

Piston types	Bowl Diameter (D_b, mm)	Bowl Depth (H_b, mm)
Heron 1	60	10
Heron 2	60	17
Heron 3	66	17

The shape of the piston top structure of this study will be selected based on the point of view of creating turbulent kinetic energy of the gas flow at the end of the compression stroke and safe during engine operation. Structuring parameters of the four-piston peaks that will be used in this study are presented in Table 3.

4. Results and Discussions

4.1. Compression Ratio Selection for Converted Engine

Diesel engines usually have a high compression ratio, and the shape of the combustion chamber depends mainly on the geometric size of the piston top, so when converting into a natural gas spark combustion engine it is necessary to study and consider decreasing the compression ratio to avoid the knocking occurrence [10].

Fig. 6 presented the effect of engine speed on the required Octane Number (ON) of six different compression ratios (ε = 10, 11, 12, 13, 14 and 15) under the same working conditions: fuel injection pressure is P_f = 1 bar and λ = 1, the head geometry of piston is flat, meanwhile the ignition timing was adjusted to the maximum brake torque (IT=MBT) and the throttle was fully opened (Throttle: WOT) to reduce losses on the intake port. Since the ON value of natural gas fuel is 130, the results obtained from the calculation have an ON value of smaller than 130 will be used to analysis.

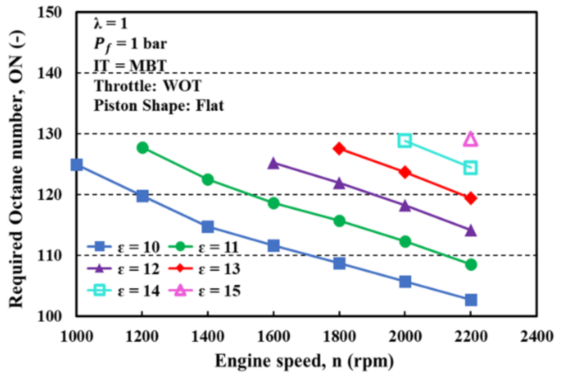

Fig. 6. Effect of engine speed on the required Octane Number (ON)

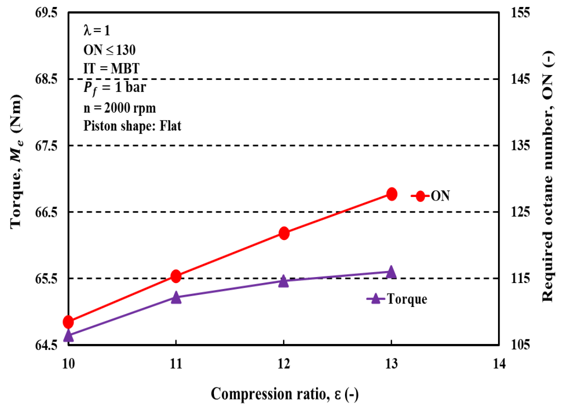

Fig. 8. Effect of compression ratio on engine torque

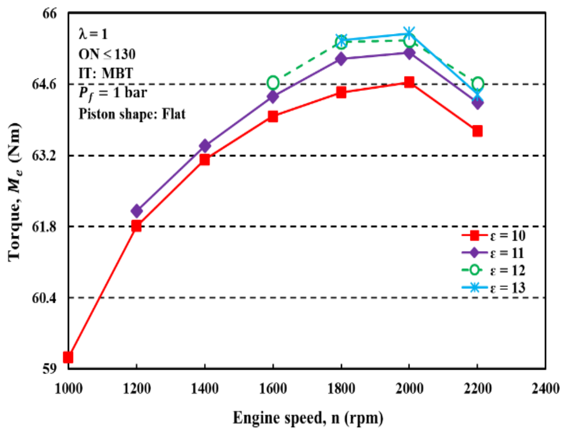

Fig. 7. Effect of engine speed on engine torque

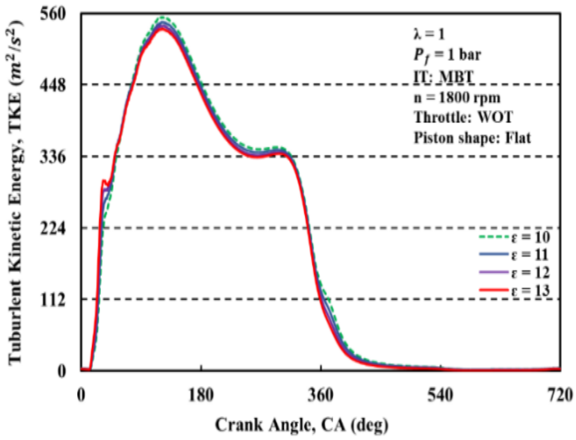

Fig. 9. Effect of compression ratio on turbulent kinetic energy

As seen in the figure, in each compression ratio, ON has the tendency to be reduced as engine speed increases. Considering the same speed, the ON value increases very quickly as the compression ratio increases, at the compression ratio is 15 on values of the engine smaller than 130 at the speed n = 2200 rpm. From these results, it can be concluded that it is necessary to reduce the compression ratio or increase the engine speed when converting diesel engines to natural gas engines.

Fig. 7 showed the change of torque according to engine speed of four different compression ratios with the value of required octan number was below the ON value of 130. Since the geometric size of the piston top does not change, changing the compression ratio will not change the shape of the combustion chamber but the gas pressure on the piston head were enhanced. Considering engine speeds in the range of n = 1000-2200 rpm, the torque of the four compression ratios tends to change relatively similarly.

When increasing the engine speed, the torque also increases and the torque reaches the greatest value at n = 2000 rpm, if the engine speed is higher, the engine torque tends to decrease. Increasing the compression ratio will improve the performance of the engine and loss more energy for compression process, in addition, in cylinder pressure also increases and this is also the cause of the increase in the knock phenomenon. Previous studies have shown that when the engine works at low speeds with a high compression ratio, it is more likely that knocking occurs than in a high-speed zone.

At the speed n = 2000 rpm the torque increases as the compression ratio increases, the cause of the increase in this case is due to increased thermal efficiency. Since the shape size of the piston top does not change, increasing the compression ratio will increase the pressure on the top of the piston without changing the shape of the combustion chamber.

The results in Fig. 8 show that ON value increases faster than torque when increasing compression ratio. That is because, when increasing the compression ratio not only increases the temperature and pressure inside the combustion chamber but also loses more the compression process.

Fig. 9 shows the effect of compression ratio on the turbulent kinetic energy in the engine cylinders of four different compression ratios. The results obtained as shown in the figure tend to change in the same cycle of the engine.

At the intake stroke corresponding to the crankshaft rotation CA = 0 to CA = 180 (deg), due to the influence of the pressure inside the engine cylinder, the TKE value of $\varepsilon = 10$ was initially smaller but then increased with the remaining three compression ratios. However, when the piston moves close to the top dead center (at the equivalent compression stroke CA = 180-360 deg), the TKE values of all four compression ratios are approximately equal as shown in the figure.

This result shows that reducing the compression ratio has increased the TKE value in the intake stroke and the first half of the compression stroke.

4.2. Effect of Piston Top Shape on Working Characteristics

Fig. 10 shows the change of engine torque when changing engine speed, at the condition as the $\varepsilon = 10$, the ignition angle adjusted to reach the maximum power (IT = MBT), $\lambda = 1$, throttle fully opened to reduce losses on the intake port. The obtained results showed that with the torque of the engine in the speed zone from $n = 1000$ (rpm) to $n = 1600$ (rpm), the Heron 1 piston top has a higher torque value than other types and when the engine speed is greater than 1600 rpm the torque value of the Heron 1 is slightly lower than the Heron 2 and Heron 3.

The reason for this difference is that the piston top shape has improved the combustion process, with different Heron styles shortening the combustion duration with the same amount of natural gas fuel inside the combustion chamber. So, the heat release rate has improved and is concentrated mainly behind the top dead center (CA = 360 deg).

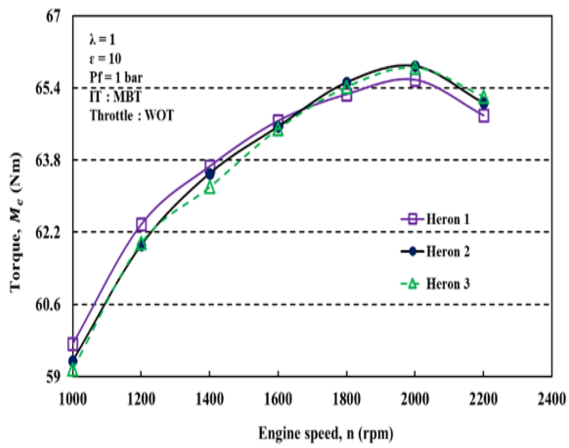

Fig. 10. Effect of piston top shape on engine torque

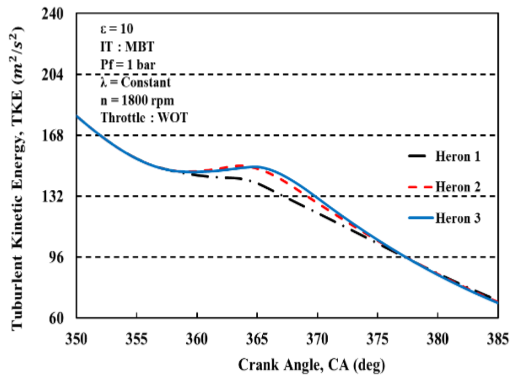

Fig. 11. Effect of bowl-in-piston on TKE as a function of crankshaft angle

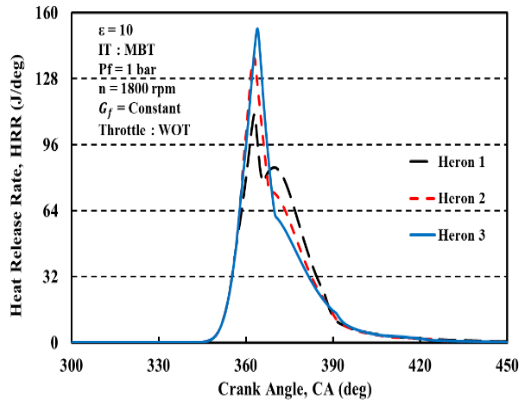

Fig. 12. Heat release rate varies with crankshaft angle

Fig. 11 presents the calculations from the data of the pressure field that varies according to the crankshaft angle of three different piston peak types. The calculation is performed at the same engine speed $n = 1800$ (rpm), $\varepsilon = 10$, fuel level pressure $P_f = 1$ bar, fully open throttle. The TKE value near the top dead center (CA = 360 deg) has been significantly improved, as seen in Fig. 6 TKE value tends to change when the volume of the bowl part on the piston top is different.

The reason is that when the piston goes up to the top dead center to the near compression stroke, there will be a squish phenomenon [11]. At that time, the air in the squish area moves with high velocity into the bowl increases the TKE, which in turn increases the ability to mix and improve the combustion process.

Observing the calculations in Fig. 12 in the same working conditions for all three Heron types we can see that the change in the fuel heat released (HRR) at a crankshaft angle is relatively similar. The rapid growth rate of HRR is concentrated in the CA = 350-360 (deg) range and the largest HRR value (Peak HRR) both appear at the back of the upper top dead center (around CA = 365 deg). This result is evidence of the hypothesis of squish appearing and directing the entire gas flow to focus on the bowl volume on the piston top. As a result, the volume of natural gas has been concentrated in the bowl volume, especially the dynamics of the gas flow in this area that has been significantly improved so that the heat is released faster [12].

Fig. 13 indicates the effect of the geometry of piston head on mass fraction burned at the same condition such as compression ratio, fuel pressure, ignition timing, λ, and engine speed was fixed in ε = 10, P_f = 1 bar, IT = MBT, λ = constant and n = 1800 rpm respectively

The mass fraction burned is a function with the variable being the crankshaft rotation angle, although the amount of fuel granted for each cycle is different, the changing trend is the same. The mass fraction burned is very compatible with the rate of heat release corresponding to the piston top types in Fig. 12. The burning rate of the Heron 3 piston is the fastest, followed by Heron 2 and Heron 1, respectively. It shows that the rate of fuel burned influences the speed of fire leading to improved fire time. The movement of the burning gas or mixture inside the cylinder increases the intensity of the turbulent and therefore during the combustion will be accompanied by some vortex. The intensity of swirling flow or turbulent kinetic energy TKE is an important indicator of flow characteristics in the cylinder, as this affects the burning rate of the fuel- air mixture. Therefore, the piston top shape will affect the mass fraction burned.

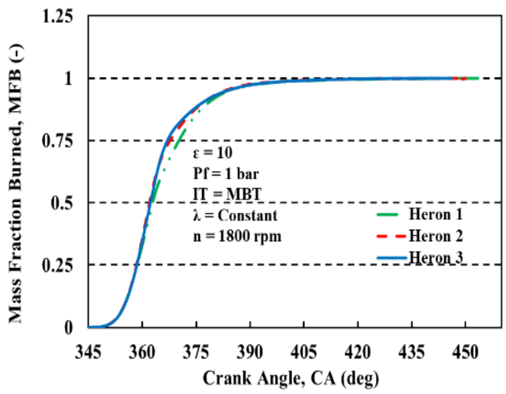

Fig. 13. Mass Fraction Burned at crankshaft angle

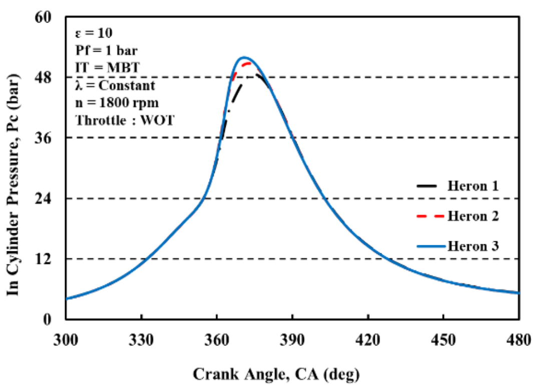

Fig. 14. Effect of piston top shape on cylinder pressure

Fig. 14 indicated in cylinder pressure changing as a function of the crankshaft angle. It could be seen that, at an engine speed of 1800 rpm, the maximum value of the pressure in the cylinder matches the HRR curves, as shown in Fig. 12. With the higher heat release rate of Heron 3, resulting a rapid increase in pressure, which leads to the maximum pressure inside the cylinder being increased. Thus, the maximum pressure inside the cylinder of Heron 3 is the maximum followed by Heron 2 and Heron 1.

The working characteristics of the internal combustion engine depend on the formation of the mixture before and during combustion. The movement of the air flow into the cylinder is the turbulent flow with the complex variation of the dynamic flow. During the loading journey, the dynamics of the air-fuel mixture increase, this value will then rapidly decrease as the piston moves towards the TDC about a third of the compression journey.

5. Conclusion

The results of the research can be drawn as following:

Engine torque tends to increase when increasing the compression ratio, however, the required ON tends to increase faster than torque, so to avoid knocking and let the engine safely work in the speed zone from 1000-2200 rpm needs to reduce the compression ratio to ε = 10 compared to the original engine.

Reducing the compression ratio helps to increase the turbulence in the intake stroke and the first half of the compression stroke, which is beneficial to the mixing process, fuel combustion, and performance.

The squish area was varied by the modification of the bowl-in-piston, thus the turbulent kinetic energy of the gas flows at the end of the compression stroke increased in comparison with the flat head piston.

Piston Heron 2 has optimized economic and technical ability when giving higher torque than other forms in most engine speed regions. Therefore, the Heron top piston is considered suitable for gaseous fuels such as CNG due to improved combustion by taking advantage of the squish phenomenon inside the cylinder.

References

[1] Amir-Hasan, Kakaee, Pourya, Rahnama, AminPaykani, Influence of fuel composition on combustion and emissions characteristics of natural gas/diesel RCCI engine, Journal of Natural Gas Science and Engineering, Volume 25, July 2015, Pages 58-65. https://doi.org /10.1016/j.jngse.2015.04.020

[2] Muhammad ImranKhan, TabassumYasmin, Abdul Shakoor, Technical overview of compressed natural gas (CNG) as a transportation fuel, Renewable and Sustainable Energy Reviews, Vol. 51, November 2015, pp. 785-797.

[3] Busch, S., Zha, K., Perini, F., Reitz, R. *et al*., Bowl Geometry Effects on Turbulent Flow Structure in a Direct Injection Diesel Engine, SAE Technical Paper 2018-01-1794, 2018.

[4] Barbouchi, Z & Bessrour, Jamel. (2021). Turbulence study in the internal combustio engine. Journal of Engineering and Technology Research Vol.1 (9), pp. 194-202, December, 2009, Available online at http://www.academicjournals.org/jetr

[5] Bhasker, Pradeep, Krishnaiah, Ravi, E, Porpatham. Krishnaiah, R., Ekambaram, P., Jayapaul, P., Investigations on the effect of Piston Squish Area on Performance and Emission Characteristics of LPG fuelled Lean Burn SI Engine, SAE Technical Paper 2016-28-0123, 2016.
https://doi.org/10.4271/2016-28-0123

[6] Chauhan Bhupendra Singh & Cho, Haeng-Muk., A study on experiment of CNG as a clean fuel for automobiles in korea, Journal of Korean Society for Atmospheric Environment, 2010
https://doi.org/10.5572/kosae.2010.26.5.469

[7] Dass, Jeevan, Lakshminarayanan, P A., Conversion of Diesel Engines for CNG Fuel Operation. In book: Design and Development of Heavy Duty Diesel Engines, 2020, pp.341-392
https://doi.org/ 10.1007/978-981-15-0970-4_9

[8] James Sevik, Michael Pamminger, Thomas Wallner, Riccardo Scarcelli, Steven Wooldridge, Brad Boyer, Scott Miers, Carrie Hall, Influence of charge motion and compression ratio on the performance of a combustion concept employing in-cylinder gasoline and natural gas blending, Journal of Engineering for Gas Turbines and Power, December 2018, Vol. 140/ 121501-1
https://doi.org/10.1115/1.4040090

[9] Krishna, R S., Conversion of diesel engine to CNG engine of commercial vehicles and emission control, International Journal of Mechanical and Production Engineering, ISSN(p): 2320-2092, ISSN(e): 2321-2071 Volume- 6, Issue-11, Nov.-2018,
https://doi.org/10.13140/rg.2.2.34701.49125

[10] Lee, K.H., Lee, C.S., Effects of tumble and swirl flows on turbulence scale near top dead centre in a four-valve spark ignition engine, Proceedings of the Institution of Mechanical Engineers Part D, Journal of Automobile Engineering, January 2005, 217(7), pp. 607-615
https://doi.org/10.1243/095440703322114988

[11] Muhammad ImranKhan, Tabassam Yasmeen, Muhamma Ijaz Khan, Muhammad Farooq, Muhammad Wakeel, Research progress in the development of natural gas as fuel for road vehicles: A bibliographic review (1991–2016), Renewable and Sustainable Energy Reviews, Vol. 66, December 2016, pp. 702-741.
https://doi.org/10.1016/j.rser.2016.08.041

[12] Young-Wook Chin, Ronald Douglas Matthews, Steven P. Nichols, Thomas M. Kiehne, Use of fractal geometry to model turbulent combustion in SI engines, Combustion Science and Technology, 1992, Vol. 86, no. 1-6, pp. 1-30.
https://doi.org/10.1080/00102209208947185