Amelioration of Combustion of Hydrogen Rotary Engine

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ABSTRACT: Hydrogen rotary engines are promising power sources for the future hydrogen society, and improving their thermal efficiency helps propel tomorrow's car farther. This research looked into the combustion process to reduce fuel consumption. Primary studies of combustion duration and location showed that increasing the local mass burning rate in the L-side of the combustion chamber by generating turbulence was beneficial. Experiments with optical and real rotary engines proved that turbulence effectively increased mass burning rate in L-side and consequently engine's thermal efficiency and its lean limit.

KEY WORDS: (Standardized) heat engine, rotary engine, combustion analysis, gas flow, (Free) Hydrogen, Turbulence [A1]

1. INTRODUCTION

The modern society regards the car as both a necessity and a polluter. From one side the car has become a very important means of transportation, giving the users the freedom to cheaply travel several hundred kilometers at once. But on the other side, the emissions from the fossil-fuel-burning internal combustion engine polluted the air and participated in the global warming. Hydrogen in this context is sought after for it can be burnt in the combustion chamber of engines, such as hydrogen Rotary Engines (RE) without the carbon dioxide emissions.

The objective of this research was to assess the effect of changing the mass burning rate and the location of combustion on the efficiency of hydrogen RE.

At first, in a preliminary study, the change in efficiency due to enhancing combustion through increasing mass burning rate was assessed using simple 0D calculation. Moreover, the outcome of changing the location of enhanced combustion was evaluated by modifying the spark plug placement and number.

Turbulence, a known combustion enhancer, was then used as a tool to achieve higher mass burning rate. 3D flow simulation showed that an alteration to the shape of combustion chamber increased turbulence in the desired location and an optical engine test showed that this turbulence effectively increased mass burning rate.

Finally the modified combustion chamber was adopted into a real engine and its results were compared to those of unmodified hydrogen RE.

2. CHARACTERISTICS OF RE AND HYDROGEN

2.1. Combustion in RE

In RE, the combustion chamber is formed between the rotor and the trochoidal rotor housing (Fig. 1). As the eccentric shaft (ECC) rotates, the rotor moves and accordingly, the volume between the rotor and the rotor housing changes creating intake, compression, expansion and exhaust strokes. The long and narrow combustion chamber has two important effects: from one side it generates a higher surface to volume ratio (S/V ratio) than typical reciprocating engine. From the other side, it gives the location of the combustion a larger influence on the engine output and thermal efficiency. The specifications of hydrogen RE operated in this study are summarized in Table 1.

Table 1 Specification of this research’s engine

| Engine Model  | Displacement | Compression Ratio |
|---------------|--------------|-------------------|
| 13B           | 0.654 L per rotor | 10.0:1 |

Fig. 1 RE and its combustion chamber
Furthermore, the combustion chamber changes in shape during compression and expansion strokes (Fig. 2). The volume of the Trailing side (T-side) decreases and that of the Leading side (L-side) increases with rotation; creating a strong one-way flow from T to L-side which can sometimes exceed 40m/s even at 1500 rpm. This characteristic one-way flow plays a very important role in shaping the combustion of RE. The speed of the flow is typically larger than the burning velocity, which prevents the flame from advancing against the flow.

The combustion inside RE is therefore divided into three distinct phases: main combustion, squish combustion and after-burn. The three phases of combustion are shown in the heat release rate in Fig. 3. During the main combustion, the flame propagates mainly in the L-side of the combustion chamber. This phase usually releases largest amount of heat from fuel and it is where the highest pressure increase is observed. When the mixture in L-side is partially consumed, the combustion is maintained by the flow of mixture from the T-side. This phase is called the squish combustion and it is characterized by being steady: the heat release rate in this combustion depends on the physical flow of mixture rather than on the chemical characteristics of the fuel. Finally the third phase is called the after-burn; it is when the fuel at the edge of the combustion chamber burns. The heat release in this phase is usually very slow and its contribution to thermal efficiency is minimal.

Past researches have concentrated chiefly on reducing the late combustion to ameliorate the thermal efficiency of gasoline RE. The idea was that reducing heat release during both squish combustion and after-burn would reduce exhaust losses. Okui et al. 2009\(^{(1)}\) for example, were able to improve the Indicated Mean Effective Pressure (IMEP) by more than 4% by simply moving the T-plug position farther away from the short axis.

The effect of total mass burning rate in the combustion chamber on RE running with hydrogen remains to be observed.

### 2.2. Combustion of hydrogen

The characteristics of hydrogen combustion differ from that of gasoline in three main areas: burning velocity, quenching gap and flammability limit. Table 2 compares between the two fuels. Inside the engine, the smaller quenching gap of hydrogen means that flame propagates closer to the walls of the combustion chamber; and therefore cooling losses are larger. Moreover, the higher burning velocity of hydrogen means that heat release rate is bigger leading to higher temperature and accordingly larger cooling losses too.

To overcome this problem, engine developers benefited from the ability of hydrogen to burn under very lean conditions. Hydrogen engines are therefore typically operated with lean mixtures\(^{(2)}\). From one side, the lean operation increases the mass of working gas to be heated through more air intake at fixed amount of fuel injection; and thus reduces the peak temperature. From the other side, burning velocity of lean mixture is slower; which prevents excessive heat release rates and accordingly reduces peak temperature. On the negative side however, the lower heat release rates generate larger exhaust losses. For this reason, researchers in hydrogen engines started looking for increasing total mass burning rate under lean conditions.

| Characteristics of combustion of Hydrogen and gasoline | Hydrogen | Gasoline |
|--------------------------------------------------------|----------|----------|
| Burning velocity at stoichiometric conditions | 3.42 m/s | 0.47 m/s |
| Quenching gap | 0.6 mm | 2.0 mm |
| Leaner flammability limit at atmospheric conditions | \(\lambda = 7.4\) | \(\lambda = 1.8\) |

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\(\textit{Fig. 2: Changes in shape of combustion chamber during compression and expansion strokes.}\)

\(\textit{Fig. 3 Heat release rate showing the phases of combustion of RE}\)
3. PRELIMINARY STUDY

3.1. Calculation assuming uniform combustion

As mentioned earlier, hydrogen’s narrow quenching gap causes higher cooling losses. Moreover, RE is characterized with a large S/V ratio, which makes it more prone to cooling losses than reciprocating engines. Amelioration to engine output due to larger total mass burning rate was therefore doubtful. A zero-dimensional calculation model was therefore developed to estimate the effect of increasing total mass burning rate on indicated efficiency.

3.1.1. 0D-calculation model

The model worked as follows:

1. Using the Wiebe function, the shape of real heat release rate curve was drawn assuming that all the fuel is burnt without losses, giving the heat release rate at each eccentric shaft angle.

2. For each eccentric shaft angle, the volume of combustion chamber was considered constant and adiabatic compression and expansion, isochoric heating and ideal gas law were used to calculate the pressure and temperature.

3. Heat transfer through the walls was calculated using Woschni formula, and cooling loss at each angle was estimated.

4. The apparent heat release rate was calculated by subtracting cooling losses from real heat release rate.

The data of the current engine was used to calibrate the model. The engine was run at 1500 rpm and wide open throttle; excess air factor \( \lambda \) was adjusted to 2. The model was calibrated to show the same cooling loss, exhaust loss and indicated mean effective pressure (IMEP) as those of the current engine; and its apparent heat release rate was adjusted to overlap with that of the engine.

3.2. Studying the location of enhanced combustion

As mentioned in section 2.1, the long and narrow shape of the combustion chamber of RE, along with strong one-way flow causes the flame propagation difference between T and L-sides. However in the 0D model above, the combustion was considered to be uniform. Real engine experiment was therefore used to study the effect of location of strengthened combustion, by varying the location and number of spark plugs, supplementing the findings of Okui et al 2009(1).

In Fig. 6, the results of three cases were shown: “Standard” was the standard engine with one plug at T-side and one plug at L-side. “2@T-side” meant that there were two plugs at the T-side, and their position was more advanced in the T-side direction than the standard. “2@L-side” meant that two plugs were located in L-side of combustion chamber.

It was found that the 2@T-side reduced the squish combustion, the heat release rate curve after 45° was lower than the others. This meant that the combustion at T-side became faster and accordingly the combustion duration became shorter. On the other hand, having two plugs on the L-side increased the heat release rate early in the expansion stroke; this meant that the flame propagation speed increased, ameliorating the local mass burning rate in the L-side. In terms of output, the 2@L-side configuration improved fuel consumption by 1.06% as compared to Standard, while that of 2@T-side reduced it by 0.54%.

3.1.2. Comparing short and slow combustions

To calculate the short combustion, the following assumptions were taken:

1. The total amount of heat release from fuel was equal to current engine.
2. Start of combustion was similar to current engine.
3. All the heat release from fuel was forced to end at a specified ECC angle.

Figure 4 shows the apparent heat release rate (after subtracting cooling losses) from the current engine and from the short combustion where heat release was forced to end at 30°. Figure 5 shows the increase in IMEP and cooling losses as well as the decrease in exhaust losses due to the calculated shorter combustion duration as compared to the current engine. It was observed that as the combustion duration decreased, the reduction in exhaust loss became larger than the increase in cooling loss; ultimately increasing the IMEP.

This 0D calculation therefore proved that increasing total mass burning rate had a good potential to ameliorate the indicated efficiency of the engine in the lean operation mode.
4. INCREASING TURBULENCE IN COMBUSTION CHAMBER

Creating turbulence inside the combustion chamber is a useful means to increase the flame propagation speed, and researchers of reciprocating engines tried to modify air intake routes and injection patterns for this purpose. RE, however, benefits from the strong one-way flow from T-side to L-side. Increasing turbulence therefore can happen by modifying the shape of the combustion chamber.

4.1. Modifying combustion chamber

As described above, RE is characterized by a strong one-way flow from T-side to L-side. In the current engines, with a smooth Middle Deep Recess (MDR), the flow is chiefly unidirectional and laminar-like. To create turbulence, a blade was added in the recess. The upstream laminar-like flow would hit the blade and turn into turbulent flow downstream of the blade.

This configuration was called Turbulence Generating Recess (TGR), and it was chosen for two main reasons. From one side, only the downstream became turbulent, and the location at which the local mass burning rate was increased through turbulence could then be adjusted. From another side, the blade could be added to the current rotor with minimal modifications restricting the differences between MDR and TGR engines to turbulence levels only, because the change in compression ratio due to TGR was less than 0.1 and was considered insignificant. Figure 7 shows the MDR and TGR layouts.

4.2. Calculation of turbulence energy

The objective of TGR was to increase turbulence, so the first analysis was that of turbulence energy. Turbulence energy is a term used to describe the kinetic energy of that component of fluid flow which represents a departure from the average kinetic energy of the fluid (also known as eddy kinetic energy).

For each mesh point, turbulence energy (TE) was calculated using equation (1). \( u'^2 \), \( v'^2 \) and \( w'^2 \) represent the variance of velocities of all the particles that passed through that point at a given time in three dimensions x, y and z: x-axis being parallel to the short axis, z-axis parallel to the center line of the eccentric shaft and y-axis perpendicular to the (x,z) plane.

\[
TE = \frac{1}{2} \left( u'^2 + v'^2 + w'^2 \right) 
\]

Eq. (1)

The turbulence energy was calculated for each degree of rotation of eccentric shaft between -30° to +120° ATDC, at the conditions shown in Table 3. Only the mesh points on the plane passing by the center of the recess parallel to x and y directions were considered. For each unit of the y-axis, the average of turbulence energy along the x-axis was calculated.
Table 3 Engine conditions for turbulence calculations

| Speed  | 820 rpm |
|--------|---------|
| Intake pressure (In.P.) | -50 kPa (gauge) |
| Ignition | No Ignition |

In Fig. 8, the vertical axis represents the distance along the y-axis of the engine, with the short axis being the zero reference and the T-direction taken as the positive one. The horizontal axis in Fig. 8 represents the ECC angle. A gray scale color code was used to show the turbulence energy, with darker color indicating higher turbulence. The trajectories of rotor ends (apex), recess ends and TGR blade (in case of TGR) were also drawn.

Figure 8 (a) shows that turbulence energy of MDR was low; this meant that the one-way flow was unidirectional and laminar-like during the whole calculation period. In the case of TGR however, Fig. 8 (b) shows that the unidirectional flow became turbulent at the level of the blade and downstream in the L-direction.

It remained therefore to be clarified whether the turbulence of TGR would actually enhance the combustion. There was therefore a need to look directly at the combustion process.

4.3. Optical engine experiment with hydrogen

In order to observe the effects of TGR on the combustion, an optical engine with a window on the combustion chamber was used. The general description of the engine was presented in Fig. 9. The combustion pictures were taken with a high-speed camera Phantom 7.3, which was triggered by the engine control unit. The axis of the camera was parallel to the z-axis, and consequently the pictures of the combustion were perpendicular to the same z-axis.

The optical engine was a one rotor version of the engine employed in the previous sections, with the rotor being equipped with two different recesses: MDR and TGR. Pressure transducers incorporated in the spark plugs were used to monitor conditions inside the combustion chamber. The engine operating conditions were stated in Table 4.

For each ECC angle and for each unit of the y-axis, the average luminance along the x-axis was calculated.

Table 4 Conditions of optical engine experiment

| Speed  | 1500 rpm |
|--------|---------|
| In.P.  | 0 kPa (gauge) |
| \( \lambda \) | 2.1 |
| Ignition timing | -5° ATDC |
Figure 10 shows the luminance that was generated by the combustion when the engine was run with MDR (Fig. 10 (a)) and TGR (Fig. 10 (b)). The calculated average luminance was represented using a gray scale color code, where darker color corresponded to higher luminance. Figure 10 was drawn in the same way as Fig. 8. Moreover, the places of the spark plugs (labeled T or L-Plug) were specified. The front of strong flame initiated at L-plug and in the T-direction was indicated by a white dotted line. It should be noted that the flame initiated at T-plug was not significantly changed by TGR because it was upstream in the T-direction away from the blade that generated the turbulence.

In the case of MDR in Fig. 10 (a), the front of strong flame was initiated at the L-plug and propagated in the L-direction mainly. In the T-direction, it remained parallel to the short axis. In the case of TGR, however, the front of strong flame started at the L-plug, and then advanced in both T and L-directions. In T-direction, the propagation of the strong flame showed two phases, at first it advanced towards the short axis, and secondly it retreated towards the L-side (the white dotted line in Fig. 10 (b)). To explain this phenomenon, comparison between Fig. 10 (b) and Fig. 8 (b) was necessary. It was found that the high luminance coincided with the high turbulence zone, which meant that the turbulence was effectively enhancing the local mass burning rate. This allowed the flame to propagate towards the short axis, but also to stop there since it was the limit of the turbulence zone.

In the L-direction, the TGR helped the flame reach the rotor end (apex) at about 30° while in the case of MDR the flame reached the apex at about 50°. It was important to see at this stage how this faster combustion was affecting the total heat release inside the engine.

In Fig. 11, the heat release rates of MDR and TGR were shown. The main combustion phase of MDR ended at about 50°, the same eccentric shaft angle at which the flame reached the L-apex. The TGR ended the main combustion phase earlier than MDR at 30°, which also coincided with the eccentric shaft angle when the flame reached the apex.

This meant that the TGR was successful in effectively increasing the flame propagation speed and accordingly the local mass burning rate at the L-side. It finished the main combustion and started the squish combustion earlier than MDR.

The effect of TGR on the performance of real engine remained to be clarified.

5. REAL ENGINE EXPERIMENT

5.1. Experimental setup

Two similar engines were prepared to test the effect of TGR on engine performance. One engine was equipped with MDR rotors, that is an unmodified engine, while the other was equipped with TGR rotors. The only difference between them was the blade in the middle of the recess.
The experiment covered a wide range of engine speeds, loads, λ and ignition timings. Engine performance, fuel consumption, emission and pressure curves were recorded for each experimental condition. Heat release rates and energy balance were also calculated for each case to properly assess the effect of TGR on combustion.

5.2. Results and discussion
5.2.1. Effect of TGR on heat release
At first the heat release rates of four engine operating conditions were drawn in Fig. 12: (a) idling, (b) medium speed and medium load, (c) medium speed and high load and (d) high speed and medium load.

Fig. 12 Comparison of heat release rates of MDR (light colored) and TGR (dark) in four engine conditions:

- (a) Idling: 1000 rpm, In.P. = -48 kPa, λ = 2.2
- (b) Medium speed and load: 1500 rpm, In.P. = -35 kPa, λ = 2.1
- (c) Medium speed high load: 1500 rpm, In.P. = -1 kPa, λ = 2.0
- (d) High speed medium load: 3000 rpm In.P. = -40 kPa, λ = 2.1

Similarly to the optical engine, it was observed that in all engine conditions, the TGR decreased the main combustion duration. This also coincided with an earlier peak in heat release rates. Moreover, the TGR, expectedly, caused the squish combustion to start earlier than MDR. It was important to note that the difference in heat release rate was more pronounced in low-load lean conditions, where the original MDR combustion was already very slow.

Figure 12 shows that the effects of TGR on the combustion were similar to that in the case of the optical engine. Its effect on the engine performance remained to be measured.

5.2.2. Effect of TGR on engine performance
The engine performance, in terms of gross IMEP which included pumping work, was measured with both MDR and TGR engines running in the same operation conditions. It was found that the TGR was effectively increasing the IMEP over a wide array of operating conditions.

In Fig. 13, the IMEP of TGR and MDR were plotted against λ while the engine was running at 1000 rpm and fixed hydrogen flow at 60 NL/min. Moreover, the difference (∆IMEP = TGR – MDR) was plotted on the secondary Y axis. It was observed that the difference between the two engines increased as the mixture became leaner.

5.2.3. Energy distribution analysis
In order to properly understand how TGR is increasing the IMEP of engine, two cases with different λ were studied. The engine speed was 3000 rpm and the ignition timing was fixed to 5°BTDC. The results were summarized in Fig. 14 and Table 5.

In Table 5, the cooling loss, unburned fuel and leakage losses were grouped into one category labeled “cooling and unburned fuel losses”; it was calculated as the difference between fuel energy and apparent total heat release. The ratio of exhaust losses to apparent total heat release was also calculated as numerical comparison tool for the rate of total mass fuel burning. It was expected that the TGR would cause an increase
in cooling losses and a decrease in exhaust losses due to larger total mass burning rate; specially during the main combustion.

The experimental results showed that the ratio of exhaust losses to apparent total heat release is reduced with TGR in both cases of $\lambda$. This meant that TGR was actually advancing the center of the heat release rate curve and accordingly the theoretical efficiency of the combustion.

In terms of cooling and unburned fuel losses however, the results differed between the case of $\lambda = 1.8$ and $\lambda = 2.3$. In the case of $\lambda = 1.8$, the cooling and unburned fuel losses increased with TGR; this was attributed to the expected increase in cooling losses. However, in the case of very lean combustion ($\lambda = 2.3$), the cooling and unburned fuel losses were reduced in the case of TGR. This unexpected phenomenon was attributed to the effect of unburned fuel. The combustion of MDR in this particular case was too slow, ending before the complete burn of gases. The combustion of TGR however was fast enough to consume more of the mixture, releasing 3.7% more heat than MDR.

The TGR therefore not only increased the local mass fuel burning rate, but also helped burn fuel more completely and therefore improved the efficiency.

![Graph of heat release rates of TGR and MDR](image)

**Fig. 14** Comparison of heat release rates of TGR and MDR (3000 rpm, $H_2$ Flow = 285L/min, and IG-L = 5° BTDC)

| Table 5 Energy balance result with $\lambda = 1.8$ and $\lambda = 2.3$ |
|---------------------------------------------------------------|
| $\lambda = 1.8$ | MDR | TGR | Change | $\lambda = 2.3$ | MDR | TGR | Change |
|-----------------|-----|-----|--------|-----------------|-----|-----|--------|
| Cooling and unburned fuel losses [J/cycle]                  | 189 | 191 | 1.05%  | 196             | 177 | -9.69%|
| Ratio of exhaust loss to total heat release [%]              | 52.84 | 49.35 | -6.60% | 52.91 | 50.31 | -4.91%|
| IMEP [kPa]                                                   | 186 | 203 | 5.87%  | 204             | 228 | 11.76%|

6. CONCLUSION

This research was another step in the quest towards high efficiency hydrogen rotary engine. It focused on the optimization of combustion of current hydrogen rotary engine. 1- With 0D simulation, it was found that increasing total mass burning rate at lean conditions of engine has the potential to increase the indicated efficiency of the engine. Moreover, by varying spark plug positions and numbers, it was found that the local mass fuel burning rate in the L-side only should be increased.

2- Turbulence in the L-side of the combustion chamber, generated by a blade in the recess, was found effective at increasing the flame propagation speed and therefore local mass fuel burn in optical engine.

3- In real engine, turbulence generating recess increased the IMEP of the engine by more than 10% (3000rpm, $H_2$ Flow = 285L/min). Moreover, it proved able at reducing unburned fuel and increasing lean limit of engine.

The findings of this research helped the authors get a better understanding of the combustion mechanism of hydrogen RE and improve the thermal efficiency of engine by controlling the combustion.

REFERENCES

(1) Okui, N., Takahashi, Y., Kagawa, R., Tabata, M., Moriue, O., Murase, E., Ignition and Combustion of Rotary Engine- Effect of spark-plug arrangement on flame propagation-JSAE 20085875 (2008) (in Japanese)

Review of Automotive Engineering, 30 pp.379-385 (2009) (in English)

(2) Verhelst, S., Sierens, R., Versraeten, S. A critical review of experimental research on hydrogen fueled SI engines. SAE Technical Paper 2006-01-04320 (2006)

(3) Cracknell, R. F., Alcock, J. L., Rawson, J. J., Shirvill, L. C., Ungut, A. Safety considerations in retailing hydrogen. SAE Technical Paper 2002-01-1928 (2002)

(4) Shudo T., Nabetani S., Nakajima Y. Analysis of the degree of constant volume and cooling loss in a spark ignition engine fueled with hydrogen. International Journal of Engine Research Vol 2 No 1 pp. 81-92 (2001)