Mathematical model and simulated power performance of a novel O-type engine

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Abstract. Reciprocating engines such as V-type, L-type and W-type engines are used widely in our daily life. However, there is still some room for the improvement of power performance if the configuration of the cylinder and connecting rod system is changed. In this paper, a novel O-type engine is put forward firstly. The structural features of the O-type cylinder sets are described briefly, and then the relevant mathematical model with hypotheses is established to conduct the performance analysis. Secondly, the O-type engine is compared with the general range of power performance of gasoline engine to demonstrate its merits. Then, the power performance indicators, namely the mean indicated pressure (PMI), space occupation efficiency (SOE) and heat efficiency (HE) of the novel engine are analysed and discussed in detail through MATLAB with the verification and validation which are conducted to ensure the reliability of results. And the results show that the power performance of the O-type engine is better than most of gasoline engines; the power performance indicators in the research are not monotonous at different volumes with the bore stroke ratio increasing; the proper range of the volume and the bore stroke ratio is obtained to make the engine achieve a high power performance. All in all, the research is helpful for the development and application of the O-type engine.

1. Introduction

Today, reciprocating engines, including Horizontal Engines, V-type Engines and W-type Engines and so on, have been widely used in the automobile industry [1-2]. They form the internal heat flow circuit, involving air intake, compression step, combustion, stroke and air exhausting, And the force generated during the stroke is finally applied to the components that require the power, transforming chemical energy into mechanical energy.

Daoud and Friedrich focused their attention on a novel compounding mechanism in the multi-cylinder Franchot engine and removed the rotational components, which made the engine more compact, efficient and economical [3]. And a linear controller system which proved that the performance of the engine with a linear system was better than non-linear classic controller system was designed by Hector, Carlos and Muditha [4]. Furthermore, Mahmood and Mohamad researched the beta-type Stirling engine and found that the length of regenerator would affect the energy efficiency differently, since various working gases, helix and hydrogen and the size of regenerator were capable of influencing the thermal and frictional losses [5].

By means of automotive engine bench test, Xu and Hao established a model based on OBD system constructed in BP neural network, which had a relatively good accuracy to estimate the engine torque [6]. Through the transient Strensith Analysis of Engine Block, Hu, Deng et al. provided a reliable basis for the design of dynamic structural strength of the block [7]. For the issues of Global environment and energy, Hong and Ouyang briefly introduced a mean value model of a spark-ignition gasoline engine and proposed an on-line torque estimation algorithm [8]. Based on BASIC language, Wang used the...
formula, which was used for calculating the power correction system of automobile engines and for calculating saturated vapour pressure, to write a program for calculation of automobile engine power calibration coefficient [9].

Additionally, Ebrahimi found that the dual-Millard cycle had the most work output among other thermodynamic cycles with the same environmental conditions [10]. Notay et al. studied the effects of the degradation of lubricant and discovered that it would increase the thickness of piston ring pack [11]. Tormos et al. built a model to gain the mechanical losses and the consumption of auxiliary energy, and discussed the promising application of the model, which would do great favour to reduce the emission of greenhouse gases [12]. Östman and Toivonen reduced torsional vibration dramatically by using a cylinder balancing method [13]. The investigations of the systems, vibration and many other features in the internal combustion engine have been shown to be prosperous.

Most of the prior studies focus on the improvement of some adverse phenomenon or other aspects such as vibration and ignition system, but the change of cylinder shape and connecting rod system is not focused on, such as O-type engine. In this paper, a brief introduction to the structural features of the O-type engine is displayed in section 2. Some hypotheses are introduced to control variables and simplify the calculations, and all the crucial performances are defined and displayed with the equations in section 3. In section 4.1, the necessary relevant parameters are pre-set before a comparison of PMI and HE between the O-type engine at a certain volume and general gasoline engine is carried out to show the superiority of the novel engine. Then in section 4.2, the results of simulated power performance indicators including PMI, HE and the SOE and the key structure indicators are displayed. The zero-dimension model is used to simulate the combustion process, as it is relatively simple and convenient, compared with some of the other models [14]. In section 5, the verification and validation are conducted before the results are discussed to define the high-performance range. In section 6, some conclusions are drawn that the power performance of the O-type engine could transcend or be comparable to that of general gasoline engine; the power performance indicators in the research rise first and then drop at different volumes with the bore stroke ratio increasing; when the working range k is set to 60°, the proper range of the volume and the bore stroke ratio is acquired to make the engine achieve a high power performance.

2. Brief construction of a novel O-type engine

The operating principle of O-type engine is similar to that of other reciprocating engines, which use combustion to generate energy for pistons to perform reciprocating motion. The main differences are the arrangement of the cylinder and guide bar mechanism shown in Figure 1 and Figure 2.

2.1. Configuration

As shown in Figure 3 and Figure 4 below, the ring rod connects two piston rods in the same plane. Moreover, one end of the guide bar is fixed by bolts and nuts on the ring rod to avoid the relative motion between them. Moreover, the other end is connected to the auxiliary axis with a bearing. On the guide bar, there is a groove whose side wall is tangent to a cylindrical pin on the crank arm. And the groove length is dependent on the condition that the cylindrical pin could travel without any obstacle. Additionally, a bearing is set around the cylindrical pin to lubricate and alleviate the friction. Except for the specific details mentioned above, the other details are similar to other ordinary internal combustion reciprocating engines.

In Figure 3, 1-cylinder, 2-piston, 3-piston rod, 4-ring rod, 5-guide bar, 6-crank arm, 7-crankshaft, 8-auxiliary axis, 9-groove, 10-cylindrical pin.
2.2. Working Mechanism
The guide-bar mechanism which is applied in the O-type engine, is shown in Figure 2. The guide bar is
the driving part while the crankshaft is the driven member. The energy released from gas combustion
directly drives the piston and ring rod and hereafter powers the guide bar. The cylindrical pin on the
crank arm, with the energy transmitted, traces a circle alongside the inside of the groove on the guide
bar to rotate the crankshaft. During the operation, the transmission angle is 90° while the pressure angle
is 0°, which suggests the superior force transmission characteristics. In the operation, the guide bar,
driven by the piston connecting to the ring rod, sways to the left and right, making the crankshaft rotate.

3. Hypotheses and Methodology

3.1. Hypotheses of the basic parameters
The hypotheses are as follows: the combustion in O-type engines researched is a transient uniform
process; the combustion process of V- and O- types of engines researched are the same because we
assume the fuel used and other configurative characteristics of the two types of engines are similar.

3.2. Method
PMI indicates the work that is done by the engine in one complete cycle; HE indicates the extent of
transformation from the internal energy to the kinetic energy; SOE considering the volume of engine is
the value of the mean indicated pressure per unit value of the volume.

3.2.1. For the space occupied by the O-type engine. As shown in Figure 5, due to the fact that all the
guide bars only rotate within the range of k degrees, it is suggested that the space produced by the left
(360-k) degrees of the circle can be filled to implement other functions. Thus, the real theoretical volume
of only one set of cylinder and guide-bar system of the O-type engine (VCGSO), \( V_o \) can be shown in
the following:
Figure 5. Parameters of the O-type engine.

\[ V_0 = \frac{kR_{ocn} \pi^2 (r+t_{wall})^2}{45} + \frac{k \pi R_{vacuum} \pi^2 (r+t_{wall})}{180} \]  \hspace{1cm} (1)

\[ R_{vacuum} = R_{ocn} - r - t_{wall}. \]  \hspace{1cm} (2)

where \( r \) is the radius of the O-type engine’s cylinder, \( k \) is the working range of pistons for O-type engine, \( R_{ocn} \) is the length from centroid of cylinder to centroid of auxiliary axis, \( t_{wall} \) is the thickness of cylinder wall, and \( R_{vacuum} \) is the radius from the inside of ring rod to centroid.

3.2.2. Principles for the performances of O-types of engines. The power performance of the reciprocating internal combustion engine could be figured out as follows:

The compression ratio, \( \varepsilon \), can be shown as following,

\[ \varepsilon = \frac{v_{min}}{\frac{k^2 \pi^2 r^2 R_{ocn}}{180}} \]  \hspace{1cm} (3)

where the \( v_{min} \) is the volume of combustion chamber.

In Equation (4), \( S \) is the stroke of piston.

\[ S = \frac{k \pi R_{ocn}}{180} \]  \hspace{1cm} (4)

In Equation (5), \( Ra \) is the bore-stroke ratio.

\[ Ra = \frac{2r}{S} \]  \hspace{1cm} (5)

The final expression of the mean indicated pressure, \( P_{mi} \), is shown below,

\[ P_{mi} = \frac{P_c}{\varepsilon - 1} \left[ \frac{v_p}{n_{2-1}} \left( 1 - \frac{1}{\varepsilon^{n_{2-1}}} \right) - \frac{1}{n_{1-1}} \left( 1 - \frac{1}{\varepsilon^{n_{1-1}}} \right) \right] \]  \hspace{1cm} (6)

where \( P_c \) is the compression terminal pressure, \( v_p \) is the pressure rise ratio, \( n_1 \) is the average polytropic compressibility index and \( n_2 \) is the average variable expansion index.

\[ \eta_1 = 8.314 \frac{\alpha A T_0 P_{mi}}{H P_0 \varphi_{in}} \]  \hspace{1cm} (7)

where \( \eta_1 \) is the heat efficiency (HE), \( \varphi_{in} \) is the aeration efficiency, \( A \) is the theoretical air requirement, \( \alpha \) is excess air factor, \( T_0 \) is the ambient temperature, \( H \) is the low calorific value of fuel and \( P_0 \) is the ambient pressure. Equation (6), Equation (7) and the derivation procedure refer to Ref. [15].

\[ \eta_{op} = \frac{P_{mi}}{V_0} \]  \hspace{1cm} (8)

where \( \eta_{op} \) is the space occupation efficiency (SOE).
To calculate the maximum possible value of the crankshaft diameter $d_1$ which is shown in Figure 6, Equation (10) and Equation (11) are given.

$$d_2 = R_{\text{vacuum}} - \frac{d_1}{2}$$  \hspace{1cm} (9)

where $d_2$ is the distance between the auxiliary axis and crank shaft.

$$\frac{d_1}{2 \sin (0.5k)} = d_2$$ \hspace{1cm} (10)

$$d_1 = \frac{2 \sin(0.5k) R_{\text{vacuum}}}{1 + \sin(0.5k)}$$ \hspace{1cm} (11)

4. Results

4.1. Comparative analysis

The parameters of O-type engines are shown in Table 1 according to usual conditions. Given that the VCGSO is 0.0076 m³, the PMI and HE are put into comparison with the general range of the indicators of other reciprocating engines to embody the superiority of the O-type engine. The cylinder bore-stroke ratio within a range from 0 to 10 is set on the X-axis, while the PMI and HE are set on the Y-axis in Figure 7 and Figure 8, respectively.

Table1. Parameter defaults of general reciprocating gasoline engine.

| Parameters | values | Parameters | values |
|------------|--------|------------|--------|
| $v_{\text{min}}, \text{L}$ | 5.465 x 10⁻² | $T_{\text{out}}^a, \text{K}$ | 1045 |
| $\Delta T^b, \text{K}$ | 12 | $\alpha_c^c$ | 0.004 |
| $\alpha_M^c$ | 0.85 | $M_{\text{wfr}}^d$ | 114 |
| $\alpha_l^e$ | 0.146 | $H, \text{KJ/Kg}$ | 44100 |
| $T_0, \text{K}$ | 298 | $A$ | 0.900 |
| $P_0, \text{bar}$ | 1.013 | $\phi_{\text{out}}^e$ | 0.086 |
| $n_1$ | 1.36 | $n_2$ | 1.25 |
| $t_{\text{wall}}, \text{mm}$ | 5 |

$^a T_{\text{out}}$ is the exhaust terminal temperature.

$^b \Delta T$ is the intake heating temperature rise.

$^c \alpha_c, \alpha_M$ and $\alpha_l$ are the fuel composition of carbon, hydrogen, oxygen respectively.

$^d M_{\text{wfr}}$ is the relative molecular weight of fuel.

$^e \phi_{\text{out}}$ is the residual gas coefficient.
From Figure 7 it can be seen that there is a local maximum (approximately 13.7 bar) for all the curves. With \( k \) increasing, the local maximum moves to the left, which indicates a smaller cylinder bore stroke ratio. Also, the gradient of curve increment is much steeper than that of the decrement. In addition, most of the values of PMI transcend the general range of PMI of gasoline engine at the relatively large bore stroke ratio, indicating that the O-type engine can do more work per volume of the cylinder in each cycle in this situation. If the working range \( k \) is narrow, the bore stroke ratio should become large to exceed the general range of PMI. Nevertheless, at relatively low bore stroke ratio, some of the values of the PMI are at the high position within the general range of PMI, which are also acceptable. From Figure 8 it is self-evident that the trend of HE curve is similar to that of the PMI curve. The maximum value of HE of the O-type engine can reach approximately 0.33, which is almost in the middle of the general range of HE of gasoline engine. When the working range \( k \) is large, the values of HE with a wider range of bore stroke ratio are higher than the bottom line of the general range of HE. The more favourable HE is located where the bore stroke ratio is small and the working range \( k \) is large simultaneously, and the contrary is the case. It is also true of the PMI.

4.2. Performance analysis

In order to leave enough space for the crank arm, the working range is set to 60° for further calculations and analysis. The range of VCGSO and the bore stroke ratio are between 0.00001 m\(^3\) and 0.01001 m\(^3\) and between 0.00001 and 5.00001 respectively, which are wide enough to do the analysis.
From Figure 9, it is self-evident that the PMI increases to the maximum along the increasing volume direction, and it increases dramatically firstly and then declines slowly along the direction of the increasing bore stroke ratio. More specifically, the pressure has a global maximum value of about 14.22 bar when the bore stroke ratio is approximately 2, and the volume is near 0.01 m$^3$. In Figure 10 it shows that with the volume declining, the SOE of the O-type engine increases with a more substantial rate when the volume is small. Besides, the SOE rises firstly and then falls along the bore stroke ratio direction, although the change is subtle. From Figure 11 the trends of variation of the HE and PMI are almost identical. However, the HE reaches its maximum value of 0.33 with the values of bore stroke ratio and volume being 2 and near 0.01 respectively. In addition, the HE varies slowly for large volume and bore stroke ratio. In Figure 12 it illustrates that the crankshaft diameter increases more sharply with a smaller bore stroke ratio than with a larger one. For the area where the bore stroke ratio is large, the diameter is not sensitive to the change of the volume and bore stroke ratio. The maximum value of crankshaft diameter is about 0.9 m with the values of bore stroke ratio and volume being near 0 and 0.01 respectively, yet of course this value does not have practical meaning.

5. Discussion

5.1. Verification and validation

Verification is the process that assesses whether a computational model can accurately represent the underlying mathematical model [16]. In this paper, the maximum values are more significant compared to other values, because the highest performance usually relates to the max. values of indicators. And all the results can be calculated with different step sizes in MATLAB. Thus, the max. values of indicators at different step sizes need to be found out and seen whether they are convergent. Note that the step size should be chosen carefully as the results could be the same and unconvincing if one step size is one of the factors of another. The results are shown in Figure 13, Figure 14, Figure 15 and Figure 16.
From Figure 13 to Figure 16, within the range of step size of iteration (SSI) from 1 to 23 the results suggest that the values are becoming convergent within the decreasing SSI. For each indicator of performance, we can take the SSI of 1, which is also what has been taken above to conduct the calculation process. Thus, the accuracy of the solving process can be ensured.

Validation is the process that determines to what extent a model can accurately represent the reality of interest from the perspective of the intended use of the model [16]. The power performance of the O-type engine proposed in this paper is not absurdly high compared to the range of the performance of V-type engine, for the operating principle and configuration of the cylinders of these two types of engines are identical and we only conduct the innovation of the configuration and mechanism. As a result of that, further, the validation of the calculation in combustion process is proved after figuring out the indicators of performance of a specific V-type engine through the same calculation procedure and comparing them to the general range of performance of V-type engine. In Table 2, it can be seen that all of the calculated results are located within the general range of reciprocating gasoline engines. Additionally, in the future we will try to build a prototype to test and compare the experimental results to the computational ones.
Table 2. The configurative parameters and the results of a specific V-type engine.

| Configurative Parameters | Values | Results | Values |
|--------------------------|--------|---------|--------|
| Bore stroke ratio        | 1.24   | Calculated PMI, bar | 11.44 |
| Compression ratio        | 10     | Calculated HE, %     | 26    |
| Cylinder diameter, mm    | 92     | General range of PMI in reality, bar | 7-13 |
|                          |        | General range of HE in reality\(^a\), % | 23-35 |

\(^a\) There are only a few exceptions that are beyond the general range of HE, such as Toyota Camry with a HE of 40% and Mazda SKYACTIV X series with a HE of 48% and.

5.2. Interpretation and discussion of the results

In order to confirm the best performance range, the coordination lines of the maximum performance values at different volumes are shown in Figure 17, and maximum performance is introduced which is called the maximum line in Figure 18, Figure 19, Figure 20 and Figure 21.

![Figure 17 The maximum line of each power performance indicator.](image1)

![Figure 18. The maximum line of \(d_1\).](image2)

![Figure 19. The maximum line of the PMI.](image3)
Figure 20. The maximum line of the HE.

In Figure 17 it illustrates that the maximum PMI line at different volumes has the same coordinates as the maximum HE and SOE line. Nevertheless, the crankshaft diameter has the maximum values when the cylinder bore stroke ratio is near 0. On the maximum line from Figure 18 to Figure 21, the crankshaft diameter, PMI and HE increase with the increasing volume. When the volume is small, they will especially rise sharply. However, the slope is relatively small when the volume is larger than about 0.004 m$^3$ for the maximum PMI and HE, while for the maximum crankshaft diameter the slope is relatively large all the time. The maximum SOE line drops all the way but especially drops sharply when the volume is less than 0.002 m$^3$. With a rough estimate of the volume of the cylinder and connecting rod mechanism (0.0076 m$^3$), the SOE of the specific V-type engine is only 1508.34 bar/m$^3$. Moreover, the maximum line of the crankshaft diameter is invalid because of the situation where the bore stroke ratio is too near 0, which is inapplicable. Combined with those above and the intention of the superiority over the specific V-type engine, the proper choice of the cylinder bore stroke ratio and VCGSO can be roughly determined.

Then, the limits can be initially set. The bore stroke ratio is theoretically between 2 and 2.5 from Figure 17 as the PME, SOE and He could reach the peak at that time. Before finally defining the proper choices of the bore stroke ratio and VCGSO, some restrictions should be considered. Firstly, the crankshaft diameter is assumed to be more than about 80 mm so as to withstand the torque of the drive shaft, so that the VCGSO should be larger than approximately 0.00013 m$^3$ from Figure 18. Secondly, PMI should be better than that of the specific V-type engine, so that the volume should be more than about 0.0025 m$^3$ from Figure 19. Besides, the VCGSO should be smaller than the volume of the cylinder and connecting rod set of the V-type engine if possible. For HE, however, it is sometimes plausible to sacrifice some of it to achieve the higher performance, so that HE is not included within the limits to simplify the calculation. The proper choices of the bore stroke ratio and VCGSO within the limits mentioned above are shown in Figure 22, while the more accurate and convincing choices of the two structural parameters can be determined with other detailed limits in the future.
Figure 22. The proper range for the bore stroke and the volume.

In Figure 22 the values of the possible crankshaft diameter in the graph above are the values that can be achieved under the restrictions set above, and so are the values of the possible PMI, the SOE and HE. The overlapped bullet-like area below the dotted red line is the proper range for the choice of bore stroke ratio and volume which makes the performance indicators better than the specific engine.

6. Conclusion

Based on the work and research done by other researchers in this field and in light of the results above, it can be concluded:

- The power performance such as the PMI of the O-type engine could be superior to most of gasoline engines while the HE of the O-type engine is at the middle level within the general range of HE of gasoline engine;
- The PMI, SOE and HE rise first and then fall with an increasing bore stroke ratio at different VCGSO, and PMI and HE increase with increasing VCGSO at different bore stroke ratios;
- In the situation where the working range \( k \) is 60°, the proper range of the volume and the bore stroke ratio are obtained—a bullet-like area where the bore stroke ratio is more than about 0.1 and less than about 0.6 and the VCGSO is larger than about \( 3.1 \times 10^{-3} \) m³.

This paper focuses on the power performance of the O-type engine. However, both the equivalent stress and strain conditions on the configurative components when the mechanism is in operation and the effective power output of the O-type engine can be simulated in the future work by using finite element analysis and the calibration coefficient [9] respectively in near future. Furthermore, the other detailed aspects, such as the ignition order, cooling system, exhaust emission, the optimised solution of structure parameters and so on, could be combined with the novel configuration of O-type engine cylinder, and the engine torque could be examined with the combination of the OBD system and the BP neutral work [6] after the first prototype is built in the future. In conclusion, the O-type engine, as a result of the better power performance and relatively small occupying space and so on, is promising and likely to be modified and applied in automobile industry.
Reference

[1] Pukrabek, W.W. (2004) Engineering Fundamentals of the Internal Combustion Engine. Prentice Hall Publishing, New Jersey.
[2] Heywood, J.B. (2018) Internal Combustion Engine Fundamentals. McGraw-Hill Publishing, New York.
[3] Daoud, J.M., Friedrich, D. (2018) A novel Franchot engine design based on the balanced compounding method. Energy Conversion and Management, 169: 315-325.
[4] Bastida, H., Ugalde-Loo, C.E., Abeysekera, M. (2017) Dynamic Modelling and Control of a Reciprocating Engine. Energy Procedia, 142: 1282-1287.
[5] Chahartaghi, M., Sheykhi, M. (2018) Energy and exergy analyses of beta-type Stirling engine at different working conditions. Energy Conversion and Management, 169: 279-290.
[6] Xu, C., Hao, L.L. (2017) An estimation of the engine torque based on OBD system. http://www.cnki.com.cn/Article/CJFDTotal-SHQC201702005.htm.
[7] Hu, Y.M., Deng, Z.X., Zhu, Z.G., Wang, P., Tan, J., Zhang, J.S., Qing, H.B. (2005) Transient Strength Analysis of Engine Block. The State Key Laboratory of Mechanical Transmission, 26: 63-67.
[8] Hong, M.N., Ouyang, M.G. (2009) On-board Torque Estimation Base on Mean Value SI Engine Models. Journal of Mechanical Engineering, 45:290-294.
[9] Wang Z.P. (1993) Computerized Calculation of Automobile Engine Power Calibration Coefficient. Journal of Highway and Transportation Research and Development, 10: 47-53.
[10] Ebrahimi, R. (2018) A new design method for maximizing the work output of cycles in reciprocating internal combustion engines. Energy Conversion and Management, 172: 164-172.
[11] Notay, R. S., Priest, M., Fox, M.F. (2019) The influence of lubricant degradation on measured piston ring film thickness in a fired gasoline reciprocating engine. Tribology International, 129: 112-123.
[12] Tormos, B., Martin, J., Carreño, R., Ramírez, L. (2018) A general model to evaluate mechanical losses and auxiliary energy consumption in reciprocating internal combustion engines. Tribology International, 123: 161-179.
[13] Östman, F., Toivonen, H.T. (2008) Active torsional vibration control of reciprocating engines. Control Engineering Practice, 16: 78-88.
[14] Zhong, C.M. (2004) Research on mathematics simulation during the combustion for internal-combustion engine. http://en.cnki.com.cn/Article_en/CJFDTOTAL-KJJJ200406017.htm.
[15] Jiang, D.M. (2002) Advanced principles of internal combustion engine. Xi’an Jiaotong University Press Publishing, Xi’an.
[16] Schwer, L.E. (2006) Guide for Verification and Validation in Computational Solid Mechanics. The American Society of Mechanical Engineers Publishing, New York.