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THERMODYNAMIC ANALYSIS OF A SINGLE CYLINDER CRANK-ROCKER ENGINE

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ABSTRACT

A thermodynamic model for the simulation of a single curved-cylinder crank-rocker engine which operates on gasoline fuel is presented. The thermodynamic model is based on the single and two-zone heat release models. The general model was expanded to include friction losses as a function of engine speed and combustion efficiencies at different operating points. The model was developed in MATLAB to predict engine performance and efficiencies on a motorcycle engine. The calculated data is then used to plot various thermodynamic parameters and the engine performances with respect to crank angle. The engines volume, indicated cylinder pressure, heat release rate, pressure-volume diagram, and engine brake torque have been evaluated at different crank angle positions. The results from the simulation model were plotted and compared with existing data. It was found that for the same engine capacity, the performance of the crank-rocker engine was better than the slider-crank engine when both utilized the same injection timing.

Keywords: thermodynamic model, crank-rocker engine, slider-crank engine, single-zone model, two-zone model.

INTRODUCTION

The performance and emissions of internal combustion engines are crucial to humankind due to the depleting oil reserve and global warming. Automotive sector has contributed largely significant to the global air pollution leading to the deterioration of the environment. Therefore, Automobile companies continue to find and develop innovative technologies in order to increase the engine efficiency and reduce emissions. The optimization process to increase the ICEs efficiency is very difficult because the current ICEs almost reached its optimum limit. It is also being installed in millions of vehicle on the road today and this has led to high CO2 emission. As a result, any improvements in engine performance must incorporate the economic principles and control the emissions.

There are a large number of technologies now being developed and evaluated which can improve the internal combustion engine, for example these technologies include fuel combustion strategy, port fuel injection, direct fuel injection, turbo-charging with direction injection system, variable compression ratio engine, variable displacement engine, variable valve timing and lift and etc.

Several technological developments related to the improved engine efficiency and reduced emissions have been reported by Ford Motor Company [1]. They released a report with the address on environmental concerns. The important focus over recent years has led to various technological advances in engine control resulting in the increase in efficiency. New technologies such as variable valve timing have been produced recently in production vehicles. Reference [2], has proposed a concept called Variable Valve Actuation to predict the outcome of the current variable valve timing technology to be camless valve actuation which can lead to efficiency improvements.

Variable compression and variable displacement engines have been addressed and experimentally investigated by researchers and automobile manufacturers [3-8]. Several authors [3, 5] have proposed different design techniques to achieve variable compression ratios. Variable displacement concepts have been analysed in many different scientific publications [5, 8]. The key advantages of variable compression and variable displacement engine are reduced the fuel consumption and increase the power out.

In general, very few researches look into the possibility of utilizing different engine configuration beside the conventional slider-crank engine; despite some concepts of using new engine configuration have been developed to various extents a long time ago. Several researches have been carried out in order to convert the conventional reciprocating motion of a piston into rotational motion of a crankshaft [9-11]. Also, several automotive engineers had attempted to design engines incorporating oscillating curved pistons and some of them patented their work, e.g. L.J. Wolff 1929, Monti Farrell 1992 and Morgado 2006 [12, 13]. The engines were very powerful but there was high incidence of sealing loss which led to high emission. The proposed design was also very complex. A new configuration of an opposed-piston crank rocker engine has been proposed by [14, 15]. It was operating on a 2 and 4-stroke, and each cylinder contained two pistons placed horizontally facing head-to-head and moved in opposition direction to each other. In general, past researches have been conducted in order to improve engine performance in reducing emissions and fuel consumption. Although several concepts had been developed to various extents in the past, the progress was still slow.

In this paper, a novel crank-rocker engine has been designed and modeled. A single and two-zone heat release models was developed in MATLAB to predict engine performance and efficiencies on the crank-rocker engine. The model was prepared to include friction losses as a function of engine speed, combustion efficiencies, and temperature-dependent thermodynamic properties. In two-

zone model, the cylinder contents such as volume and mass are divided to consist of two regions (zones), burned and unburned zone.

CRANK-ROCKER ENGINE CONCEPT

The crank-rocker engine introduced in this work is an alternative engine configuration. The uniqueness of the engine geometry is expected to improve the engine thermodynamic cycles and perform better than the conventional engines.

The main concept of Crank-Rocker Engine was derived from four-bar mechanisms without any 'delayed angle' (balanced). The most unique is the 'zero-balanced angle' of the crank-rocker engine where it allows the crank to have a smooth rotary motion. This engine is designed to operate using a curve piston and the combustion cycle is similar to the OTTO cycle.

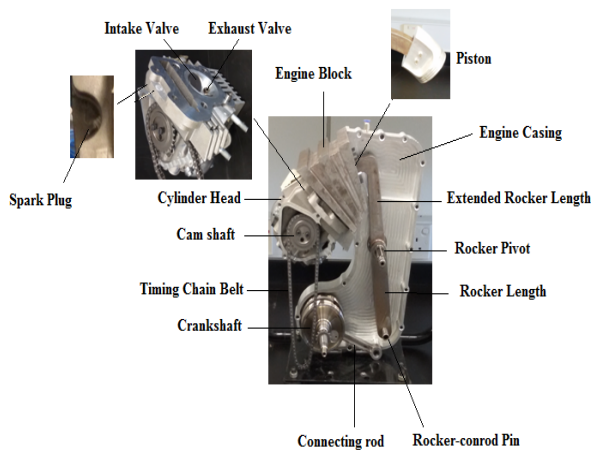

Figure-1. Single curved-cylinder crank-rocker engine.

The configuration of the crank-rocker engine as well as the basic parts and shape is shown in Figure-1. As can be seen from the figure, a curved piston assembly travels within a curved engine cylinder. The input motion comes from a combustion force acting perpendicularly at the piston which decays at the end of the rocker stroke. The time taken for the rocker to move from either extreme position to the other should be equal to the time taken for the crank to rotate 180 degrees. Similar to the slider-crank engine, the gases pressure produced from the combustion of fuel applies a direct force to the piston which is attached to the rocker tip. This force is transferred to the crank along the connecting rod, and generates useful mechanical power.

Two valves namely the intake and exhausts are attached to the head of the cylinder. A rotating camshaft mounted on the engine head (overhead cam) is used to operate the valves at appropriate time during the stroke of the piston, operated directly by push rods. The camshaft is connected to a crankshaft via a chain belt. As the crank rotates through 360^0, the rocker will oscillate through an angle θ. The engine is intended to be designed to operate using a single cylinder only for any capacity of the engine requirements in order to avoid numerous rotating parts.

THERMODYNAMIC MODEL

The thermodynamic simulation model of internal combustion engine has long been extensively used for investigating and analysing the engine performance and giving a good evaluation for new developments [16-21]. When investigating the internal combustion engine, the in-cylinder pressure has always been an important experimental parameter in automotive research and development, due to the relationship between the combustion and work processes [22]. In order to understand of how the combustion process propagates through the combustion chamber, it is important to relate all parameters such as cylinder volume change, mass loss in cylinder chamber, and the heat transfer to the cylinder pressure. All of these parameters can be obtained using the first law of thermodynamics. A single zone model has been used in order to predict the in-cylinder pressure and other parameters. A more accurate thermodynamic analysis would be to use a multi-zone model, where the cylinder is divided into a number of zones, differing in composition and properties. Each zone is uniform in composition and temperature, and the pressure is the same for all the zones [23, 24].

The model was used to predict the cylinder gas temperature and pressure, as well as the performance of the engine. In order to improve and increase the accuracy of the single-zone model, the heat transfer, variable specific heat ratios, and friction models were included. In this paper will give a more detailed data of the development of the single and two-zone models.

Modelling assumptions

Some assumptions are adopted to simplify the single-zone model equations such as follows:

1. All gases mixture is ideal gases.
2. The cylinder contents are assumed uniform throughout the cylinder.
3. The combustion is simulated as a release of heat
4. The heat released from the combustion is uniformly in cylinder.
5. The separation between burned and unburned zones is assumed to be very thin and no heat transfer between the two zones.

Single-zone model

The simplest approach is to observe the cylinder contents as a single region or zone [23, 25]. Single-zone models use the Weibe function to represent the mass fraction burned as a function of crank angle and is defined as follows [17]:

$$X_b(\theta) = 1 - \exp\left[-a\left(\frac{\theta - \theta_s}{\theta_b} \right)^{n+1} \right] \quad (1)$$

Where θ = crank angle,
θ_s = start of ignition,
θ_b = the burn duration,
n = Wiebe form factor and a = Wiebe efficiency factor.

The parameters a and n are adjustable parameters used to fit experimental data.

The cylinder gas volume, in a conventional slider-crank engine can be calculated as a function of crank angle [17, 24]:

$$V(\theta) = V_c + V_d \tag{2}$$

Where Vc is the clearance volume which is expressed as:

$$V_c = \frac{V_d}{C_r} \tag{3}$$

Where Vd is the displaced cylinder volume, and Cr is the compression ratio. The engine displacement is given by

$$V_d = \frac{\pi B^2}{4} * S \tag{4}$$

B = the cylinder bore,
S = the engine stroke,

The crank-rocker engine stroke is not like the crank-slider engine, and it does not depend on the crankshaft radius as previously used by the slider-crank engine. The expression is shown below:

$$S = 2r \tag{5}$$

The crank-rocker stroke is given by

$$S = L_{41} * \phi \tag{6}$$

Or

$$S = L_{41} * \left(2 * \sin^{-1}\left(\frac{r}{L_4} \right) \right) \tag{7}$$

Where r is the crank radius, ϕ is the rocker throw angle, L_4 is the rocker length; and L_{41} is the extended rocker length. Figure 2 shows a diagram of these figures and their relationship to engine geometry.

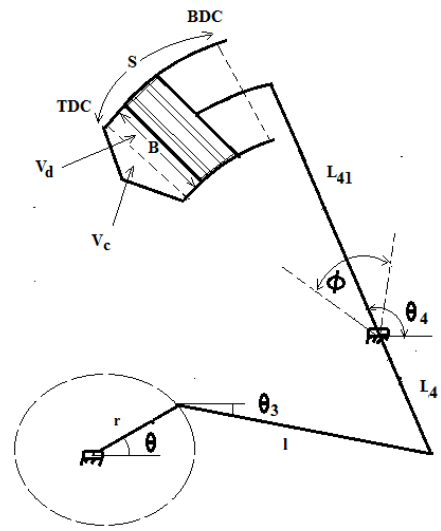

Figure-2. Crank-rocker engine geometry and variables.

The ideal gas law is defined as:

$$PV = MRT \tag{8}$$

By taking the logarithm of both sides and differentiating with respect to crank angle provides:

$$\frac{1}{P}\frac{dP}{d\theta} + \frac{1}{V}\frac{dV}{d\theta} = \frac{1}{T}\frac{dT}{d\theta} \tag{9}$$

Rearranging equation 9 and solving for dP:

$$\frac{dP}{d\theta} = -\frac{P}{V}\frac{dV}{d\theta} + \frac{P}{T}\frac{dT}{d\theta} \tag{10}$$

The first law of thermodynamics in differential form for an ideal gas with constant specific heat is given by:

$$\frac{dU}{d\theta} = mC_v\frac{dT}{d\theta} = \frac{dQ}{d\theta} - \frac{dW}{d\theta} \tag{11}$$

Where Q is the total energy,
W = the work,
dU = the change in internal energy,
Cv = the specific heat of the combustion chamber gas.

The relationship between the specific heat, universal gas constant, and the specific heat ratio γ is given by:

$$\frac{C_v}{R} = \frac{C_v}{C_p - C_v} = \frac{1}{\gamma - 1} \tag{12}$$

The net-work and heat input in the spark-ignition engine can be described form the following equations [22, 23]:

$$\frac{dW}{d\theta} = P\frac{dV}{d\theta} \tag{13}$$

$$\frac{dQ}{d\theta} = \eta_c * LHV * \left(\frac{dX_b}{d\theta}\right) - \left(\frac{dQ_w}{d\theta}\right) \tag{14}$$

When substituting equation 12 and 13 into equation 11, the instantaneous change in temperature is given by:

$$\frac{dT}{d\theta} = T(\gamma - 1)\left[\left(\frac{1}{PV}\right)\left(\frac{dQ}{d\theta}\right) + \left(\frac{1}{V}\right)\left(\frac{dV}{d\theta}\right)\right] \tag{15}$$

The heat derived from the combustion of fuel can be used in order to compute the change in the gas cylinder pressure. The heat input is expressed as [24, 25]:

$$Q_{in} = \eta_C * LHV * \left(\frac{1}{AF_{ac}}\right)\left(\frac{P}{RT}\right)V_d \tag{16}$$

Finally, the change in the cylinder pressure as a function of crank angle is defined as:

$$\frac{dP}{d\theta} = A + B + C \tag{17}$$

Where A, B, and defined as follows:

$$A = \left(-\frac{\gamma P}{V}\right)\left(\frac{dV}{d\theta}\right)$$

$$B = \left(\frac{\gamma - 1}{V}\right)Q_{in}\left(\frac{X_b}{d\theta}\right)$$

$$C = (-1)\left(\frac{1}{V}\right)\left(\frac{dQ_w}{d\theta}\right)$$

Since all of the variables have been defined, a single-zone model can be developed and established.

Heat transfer model

In general, the heat transfer from the in-cylinder gases to the cylinder wall in spark ignition engines is convection, with only a view percentage from radiation [17]. By using a Newtonian law, the heat loss to the wall is given by:

$$Q_w = h * A * (T - T_w) \tag{18}$$

Where
h = convection heat transfer coefficient
T = temperature of the cylinder gas
Tw = cylinder wall temperature
A = exposed combustion chamber surface area

Air-fuel ratio model

Generally, the air-fuel ratio can be calculated using the lambda (λ) reading, the balanced equation, and stoichiometric reaction between fuel and air. The general, balanced stoichiometric reaction for a HC fuel is defined as [25, 26]:

$$C_xH_y + a(O_2 + 3.76N_2) \rightarrow xCO_2 + \left(\frac{y}{2}\right)H_2O + a(3.76N_2) \tag{19}$$

Where x = the number of carbon atoms in the fuel,
y = the number of hydrogen atoms in the fuel,
a = constant for balance the equation.

For example, the balanced, stoichiometric reaction for C_8H_{18} fuel is expressed as:

$$C_8H_{18} + \left(\frac{25}{2}\right)(O_2 + 3.76N_2) \rightarrow 8CO_2 + 9H_2O + \left(\frac{25}{2}\right)(3.76N_2) \tag{20}$$

Upon substituting the values of x, y, and a, the stoichiometric air-fuel ratio is found to be [22, 23]:

$$AF_{stoich} = \frac{4.76(a)}{1}\left(\frac{(Molecular.weight_{air})}{(Molecular.weight_{fuel})}\right) \tag{21}$$

Finally, the actual air-fuel ratio is given by:

$$AF_{ac} = \lambda * AF_{stoich} \tag{22}$$

Friction model

It well known that friction losses influence the engine performance (indicated and brake power). Some researchers such as [17, 27] have used general linear equations to calculate FMEP losses as a function of RPM. For the spark-ignition motorcycle engine with roller bearings, the FMEP loss defined as [27]:

$$FMEP = 250(L)(RPM) \tag{23}$$

Where L is engine stroke (m), RPM is engine speed (rev/min).

Two-zone model

The two-zone model describes the burned and unburned gas areas and their association to the combustion chamber. Two-zone model can be used in order to predict the heat transfer and emissions precisely [24]. In order to develop a two-zone model, some modification of the Weibe function has to be done in order to include burned and unburned areas or zone. The burned and unburned masses can be expressed as [27]:

$$m_b(i) = m_b(i-1) + \frac{dX_b}{d\theta}(i)m_c \tag{24}$$

$$m_u(i) = m_u(i-1) + \frac{dX_b}{d\theta}(i)m_c \tag{25}$$

Where mc is the total mass in cylinder (mass of air + mass of fuel)

After computing the burned and unburned masses, the volume of burned and unburned must be obtained. Some researchers, [27] have suggested that using the polytropic relations and in-cylinder pressure-trace in order to calculate the burned and unburned volumes. The unburned volume is given by:

$$V_u(i) = \left(\frac{m_u(i)V_u(i-1)}{V_u(i-1)}\right)\left(\frac{P(i)}{P(i-1)}\right)^{\left(\frac{1}{\gamma_u(i)}\right)} \tag{26}$$

Finally, the burned volume can be obtained by [24, 27]:

$$V(i) = V_b(i) + V_u(i) \tag{27}$$

By using the ideal gas law and applying to each zone. The burned and unburned temperatures are given by:

$$T_b(i) = \frac{P(i)V_b(i)}{m_b(i)R(i)} \tag{28}$$

$$T_u(i) = \frac{P(i)V_u(i)}{m_u(i)R(i)} \tag{29}$$

The burned and unburned areas can be calculated as follows [28]:

$$A_u(i) = A(i)\left(1 - (X_b(i))^{0.5}\right) \tag{30}$$

$$A_b(i) = A(i)\left(\frac{X_b(i)}{(X_b(i))^{0.5}}\right) \tag{31}$$

Generally, instantaneous heat transfer area is calculated as:

$$A(i) = A_{ch} + \frac{\pi B^2}{2} + \pi BS(i) \tag{32}$$

RESULTS AND DISCUSSIONS

In order to analyse and display the results of the crank-rocker engine, a MATLAB code was prepared. The engine performance for a single cylinder, four stroke crank-rocker engine which operate on gasoline was calculated. The reference engine used in this research is motorcycle engine. The crank-rocker engine specification and dimensions (Table 1) used in this model are as follows:

Table-1. The crank-rocker engine main data.

Cylinder displacement	**120cc**
Cylinder Diameter	55mm
Number of Strokes	4
Number of Cylinders	1
Stroke	50.6mm
Compression Ratio	9.8
Connecting Rod	100mm
Rocker and Extended-rocker Length	138.8 and 138.1mm
Throw Angle	21^0

The engine is assumed to operate at 3000 rpm, with an equivalence ratio of 0.9. The ignition advance of the model engine is considered as 25^0 bTDC and the combustion duration is 85^0. The results for the simulation are shown in Figure-3 (a)-(e).

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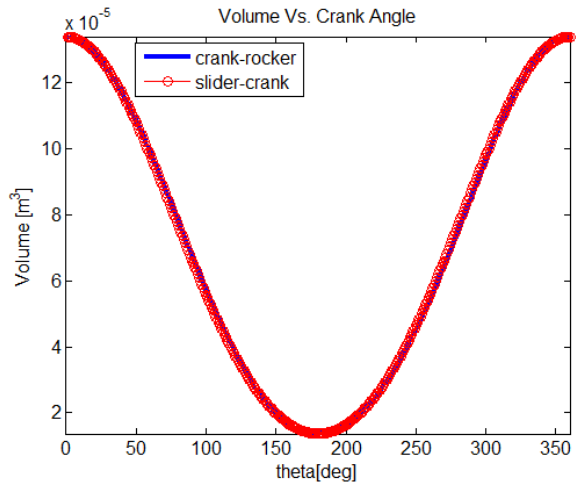

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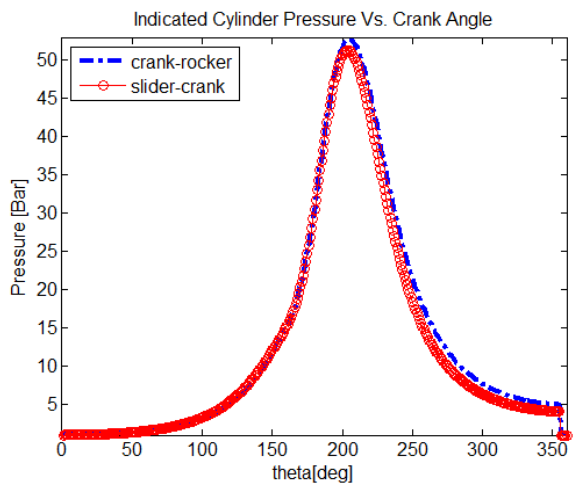

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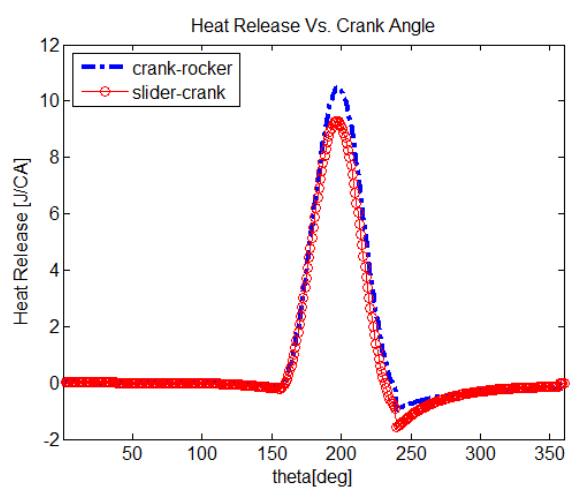

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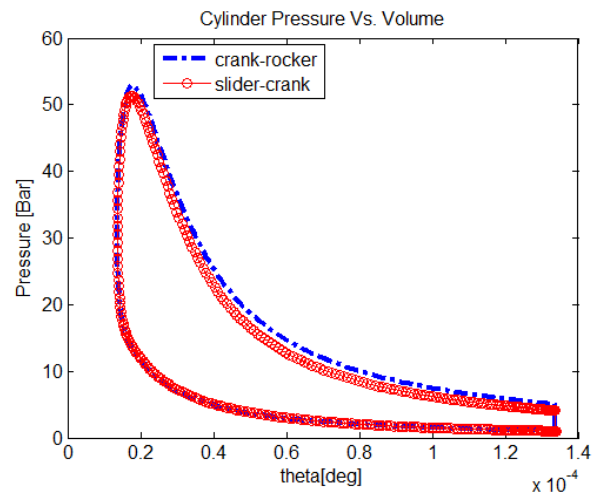

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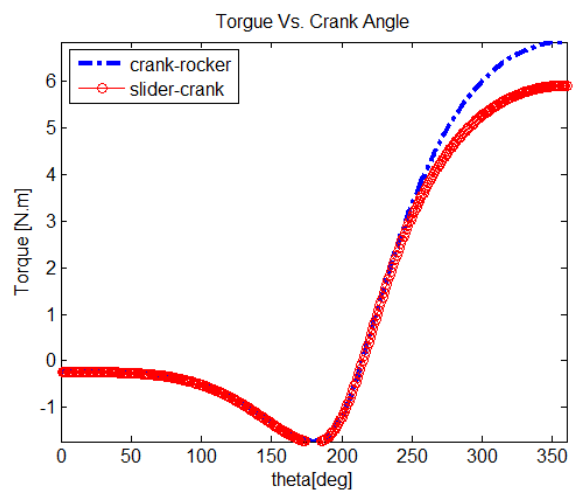

(e)

Figure-3(a-e). The model comparison between the crank-rocker (Blue line) and slider-crank (Red line) engines. (a) Engine Volume, (b) Cylinder Pressure, (c) Heat Release, (d) Pressure vs Volume, and (e) Brake Torque Vs Crank Angle.

As can be seen form Figure 3a, the two engines have the same volume. Therefore, the method for analysing the displacement of the crank-rocker was correct. Figure 3b shows in-cylinder pressure versus crank angle for both engines. At 3000 rpm engine speed, the crank-rocker engine produced higher value compared to the slider-crank engine. The peak value was 53.2 bar and accrued at 206^0 CA for crank-rocker, while the peak value of the slider-crank was 50.08 bar accrued at 203^0 CA. The explanation to this is mainly because the combustion process performance of the crank-rocker is better and the burned fuel is good and this phenomenon led to the increase in the cylinder pressure.

Heat release rate is one of the main parameters that used in internal combustion engines. Heat release pattern can explain the combustion process that occurs

inside the cylinder. Higher heat release shows that better combustion efficiency and higher NOx emission takes place in the process.

The heat release characteristics for both engines as function of crank angle is shown in Figure 3c. It is observed form Figure-3c that the crank-rocker engine produces higher heat release rate as compared to the slider crank engine due to the increase in the gas temperature and similar to those of the cylinder.

Figure-3d shows the P-V diagram for both engines. It can be noted that the configuration of the crank-rocker engine has improved the thermodynamic cycle. Because the piston motion of the crank-rocker engine moved faster in the expansion stroke, these led to the decrease in the heat transfer to the walls and hence improve the combustion process. Thus, increasing the in-cylinder pressure can produce higher work as well as higher power output (see Figure 3e).

SUMMARY AND CONCLUSIONS

In this paper, a novel crank-rocker engine model was analysed. Simulation on the basis of a single and two-zone modelling was conducted in order to obtain the cylinder pressure, P-V diagram, torque and power output. The performance of the model was compared against an existing model data for a slider-crank engine. It can be concluded that the crank-rocker engine with some optimization could improve the engine efficiency.

REFERENCES

[1] http://www.ford.com/doc/200506_sustainability_report.pdf.

[2] Ronald J. Pierik and James F. Burkhard. 2000. Design and Development of a Mechanical Valve Actuation, SAE Paper 2000-01-1221.

[3] Yamin, J.A., Dado, M.H.2004. Performance simulation of a four-stroke engine with variable stroke length and compression ratio, Applied Energy 77: 447-463.

[4] Ozcan, H., Yamin, J.A.A. 2008. Performance and emission characteristics of LPG powered four stroke SI engine under variable stroke length and compression ratio, Energy Conversion and Management 49:1193–1201

[5] Adams, W.H., Hinrichs, H.G., Pischinger, F., Adamis, P., Schumacher, W., Walzer, P. 2002. Analysis of the combustion process of a spark ignition engine with a variable compression ratio, SAE Paper 2002-01-870610.

[6] Pouliot, H.N., Delameter, W.R., Robinson, C.W. 1977. A Variable Displacement Spark Ignition Engine, SAE Paper 1977-01-770114.

[7] Freudenstein, F., Maki, E. R., Variable Displacement Piston Engine, U. S. Patent #4,270,495, 1981.

[8] Welsh, H. W., Riley, C. T., The Variable Displacement Engine, An Advanced Concept Power Plant, SAE paper 710830, 1971.

[9] Ciesio-kiewicz, A., M-yk, P. 2002. Double Pistons Internal Combustion Engine, Journal of KONES, 1(2): 56-64.

[10] Norbye, Jan P. 1972. The Wankel engine: Design Development Applications, Bailey Bros and Swinfen Ltd. Folkestone, UK.

[11] Sherman D. 2008. “The Rotary Club", Automobile Magazine, 76-79.

[12] http://www.autoblog.com/2008/02/11/oscillatingpiston-engine-that-thing-got-a-toroid-in-it/

[13] http://www.greencarcongress.com/2006/04/new_toroidal_in.html

[14] http://www.oldengine.org/members/diesel/RootesListerTS3/TS3.htm.

[15] Miros, S. 2012. New Concept of a Rocker Engine - kinematic Analysis, Journal of KONES Powertrain and Transport, 19(3): 443-449.

[16] Ferguson C. R. 1986. Internal Combustion Engines, Applied Thermosciences. New York: John Wiley and Sons.

[17] Heywood, J. B. 1988. Internal Combustion Engine Fundamentals. New York: McGraw-Hill.

[18] Sundeep Ramachandran. 2009. Rapid Thermodynamic Simulation Model of an Internal Combustion Engine on Alternate Fuels, IMES. 2: 978-988.

[19] R. Ebrahimi, B. Desmet. 2010. An experimental investigation on engine speed and cyclic dispersion in an HCCI engine Fuel, 89 (8): 2149-2156.

[20] M. Fathi, R.Kh. Saray, M.D. Checkel. 2010. Detailed approach for apparent heat release analysis in HCCI engines. Fuel, 89 (9) 2010: 2323-2330.

[21] R. Ebrahimi. 2011. Effect of specific heat ratio on heat release analysis in a spark ignition engine. Fuel, 18 (6): 1231-1236.

[22] Chun, K. M., Heywood, J. B. 1987. Estimating Heat-Release and Mass-of-Mixture Burned from Spark-Ignition Engine Pressure Data, Combustion Science and Technology, 54: 133-143.

[23] Marcus K. 2004. A Specific Heat Ratio Model and Compression Ratio Estimation. Thesis, Linköping University, Sweden.

[24] Guezennec Y. G. and Hamama W. 1999. Two-Zone Heat Release Analysis of Combustion Data and Calibration of Heat Transfer Correlation in an I. C. Engine, SAE Paper 1999-01-0218.

[25] Jeremy L. Cuddihy. 2004. A User-Friendly, Two-Zone Heat Release Model for Predicting Spark-Ignition Engine Performance and Emissions. MSc thesis, University of Idaho.

[26] Turns S. 2001. An Introduction to Combustion: Concepts and Applications, 3rd edition. McGraw-Hill Science/Engineering/Math.

[27] Blair G. P. 1999. Design and Simulation of Four Stroke Engines [R-186]. Society of Automotive Engineers Inc.

[28] Rakopoulos C. D. and Michos C. N. 2008. Development and validation of a multi-zone combustion model for performance and nitric oxide formation in syngas fueled spark ignition engine, Energy Convers. Manag, 49(10): 2924-2938.