Research Paper

A study and comparison of frictional losses in free-piston engine and crankshaft engines

Boru Jia⁎, Rikard Mikalsen, Andrew Smallbone, Anthony Paul Roskilly

Sir Joseph Swan Centre for Energy Research, Newcastle University, Newcastle upon Tyne NE1 7RU, UK

HIGHLIGHTS

• Friction mechanisms were reviewed and compared between a FPE and CSE.
• Sub-models were developed to describe friction force from friction mechanisms.
• FPE does not show advantage on piston ring friction force over the CSE.
• Total friction loss of the FPE is nearly half of the CSE.

ARTICLE INFO

Keywords:
Free-piston engine
Piston assembly friction
Friction force
Skirt friction

ABSTRACT

Friction work in free-piston engines is expected to be lower than in crankshaft engines due to the elimination of the crank mechanism. In this paper, friction mechanisms were reviewed and compared between a free-piston and crankshaft engine of similar size. The main friction mechanisms were identified to be the piston assembly including piston rings and piston skirt, valve train system, the crank and bearing system for the CSE, and the linear electric generator for the FPE. The frictional loss of each friction mechanism was estimated and discussed. A Stribeck diagram was used to simulate the piston ring friction during hydrodynamic lubrication, mixed lubrication, and boundary condition. It is found that the FPE does not show advantage on piston ring friction force over the CSE, and the frictional loss from the piston ring is even higher. While the elimination of the crankshaft system reduces the frictional loss of the FPE, and the total friction loss of the FPE is nearly half of the CSE.

1. Introduction

1.1. FPE technology

The free-piston engine (FPE) is considered a promising alternative to the conventional crank shaft engines (CSEs) [1–5]. FPEs can be divided into three categories according to piston/cylinder configuration: single piston, dual piston and opposed piston [6]. A general introduction of different FPE configurations is summarised in Table 1. The basic operation principles are equal for each concept; differences between the concepts are the number of combustion chambers and compression stroke realization [7,8].

A single piston FPE consists of a combustion chamber, and a load and rebound device. The load and rebound devices could be electric generator, gas spring chamber or hydraulic pump [9–12]. There has been successful implementation of the single piston type, coupled with a gas spring rebound chamber [13–15]. The opposed piston FPE was used almost exclusively in the early stage of the FPE development (1930–1960). It served successfully as air compressors and later as gas generators in large-scale plants [16,17]. This kind of FPE configuration essentially consists of two opposed pistons with a shared combustion chamber [18]. The dual piston (or dual combustion chamber) configuration has been topic for much of the recent research in free-piston engine technology. A number of dual piston designs have been proposed and a few prototypes have emerged, both with hydraulic and electric power output [19–26].

1.2. Reported work on FPE frictional losses

An analysis of engine friction mechanisms in four stroke spark ignition and diesel engines is presented by Heywood [27]. An approximate breakdown of rubbing and accessory friction is: piston assembly 50%; valve train 25%; crankshaft bearings 10%; accessories 15% [27]. Friction work in the FPE is expected to be lower than that of the CSE due to the elimination of the crank mechanism. Thus the friction in the wrist pin, big end, crankshaft, camshaft bearings, the valve mechanism,
gears, or pulleys and belts which drive the camshaft and engine accessories is removed [28,29]. As a result, the friction model is usually simplified during modelling of FPE systems.

Some previous reported work didn’t take the friction force into consideration in the 0/1 dimensional numerical model of the FPEs, as it was considered minimal compared with the pressure force from the cylinder, and does not influence dynamic behaviour to any great extent [29]. In order to improve the model accuracy and have some representation of frictional losses, a constant value was used to simulate the friction force during the movement of the piston with the mover of the linear electric generator, and the direction of the friction force was assumed to be opposite with the piston velocity [30–33]. The constant value was usually assumed to be 60–100 N [34].

The total friction force $F_f$ of each piston has also been estimated as a linear combination of piston velocity plus a constant $C_s$, as shown in the equation below [17]:

$$F_f = C_s v + C_i$$

(1)

$C_s$ is the kinetic friction coefficient related to the instantaneous velocity, and the $C_i$ is the static friction coefficient as a constant part of the frictional force.

Boru Jia, et al. divided the friction force of the FPEG to three components: friction from the linear electrical machine ($F_{fm}$) and friction forces between the piston rings and cylinder wall from both left ($F_l$) and right side ($F_r$) of the engine. The total friction force was written as follows [34,35]:

$$F_f = F_{fm} + F_l + F_r$$

(2)

The friction of linear electrical machine comes from the contact of the mover and the stator. It was assumed to be constant as the velocity of the piston is low. An empirical relationship was used to calculate the parameterized friction for the contact between rings and cylinder wall [19], which was written as in Eq. (3). The frictional loss of the FPE was reported to be around 5% of the indicated power from the simulation results, while it was approximately 10% of the indicated power for the CSE [34].

$$F_{prog/torque} = f \left[ -\text{sign}(v) \cdot A_f \cdot \sqrt{|v|} \right] \left[ 1 - B_f \frac{\partial \theta}{\partial t} \right] \left[ 1 + K_p \frac{p(t)}{p_0} \right] \left( \frac{d}{d_0} \right)$$

(3)

In Eq. (3), $f$ is the overall scaling factor (-); $v$ is axial velocity of piston (m/s); $A_f$ is friction parameter (-); $B_f$ is friction parameter (-); $E$ is the average temperature of lubrication oil at liner (°C); $d$ is cylinder diameter (mm); $p$ is instantaneous in-cylinder pressure (bar); $\theta_0$ is reference temperature (°C); $p_0$ is reference pressure (1 bar); $d_0$ is reference cylinder diameter (165 mm).

Chenheng Yuan undertook an investigation on the friction characteristics of piston rings in a free-piston engine generator [36,37]. The pressure distribution and thickness of the oil film were described by the Reynolds equation. The detailed effects of the piston dynamics on friction force and lubrication were studied and compared with that of a CSE. It was found that the average value of the friction force, frictional loss of the piston rings in the FPE were less than that of the CSE due to a better lubrication around the dead centre. The friction loss advantage of piston rings were reported to be not as obvious as expected [36].

1.3. Aims and methodologies

The FPEs have been studied and developed over a long time. One of the most common motivations for continued work on the topic is the purported friction advantages due to the lack of crankshaft. However, very few studies have actually calculated the friction of FPEs. This paper is an attempt to fill in this missing information. In this paper, all the engine friction mechanisms will be reviewed and compared with a CSE of similar size. Detailed sub-models will be developed to describe the friction force from the piston assembly friction, the piston skirt friction, as well as the bearing friction. The frictional loss of the two engines will be compared to give a better understanding of the lubrication condition and efficiency of them.

2. FPE technology and fundamental characteristics

2.1. FPE configuration

The FPE configuration adopted in this research is shown in Fig. 1, and comprises a combustion cylinder, a bounce chamber cylinder, and a permanent magnet linear electric machine. The two pistons are connected with the mover of the linear electric machine, and this assembly forms the only significant moving part of the system. The piston movement is influenced by the instantaneous balance of cylinder gas pressure forces, electric machine force, and frictional forces. More details of this FPE configuration can be found in our previous reports [11,31,38]. The real time displacement data of the piston is from the position sensor integrated with the linear electric machine, which is also used for proving position feedback to the linear electric machine. The FPE specifications are shown in Table 2.

The engine is operated on a turbocharged two-stroke diesel cycle, the gas exchange process is performed through scavenging ports and exhaust poppet valves in the cylinder head. The amount of gas trapped in the bounce chamber is regulated by pressure control valves, which affect the corresponding piston movement. The linear electric machine is operated as a motor during the engine cold start-up process, and then switched to an alternator to generate electricity. Successful implementations of engine cold start-up process using the linear electric machine have been reported [2,39]. Detailed modelling of the FPE system has been published in previous papers, as well as the model validation results. For more information on the numerical model of each sub-system, and detailed simulation results, please refer to Ref. [11,31,34].

2.2. Dynamic and thermodynamic difference with CSE

The piston dynamics of the FPE and cylinder pressure are shown in Figs. 2 and 3 respectively, with that of a CSE (with a crankshaft radius to connecting rod length ratio of 0.3) compared in the same figure. It can be seen that the piston movement near TDC is faster for the FPE due to the elimination of the mechanical connection system, while the peak velocity is lower. The peak cylinder pressure of the FPE is approximately the same as that of the CSE. As the friction force is likely to be influenced by the piston velocity as well as the cylinder pressure, the friction force of the FPE is expected to differ from that of the CSE due to the differences shown in Figs. 2 and 3.
The changing trends of cylinder pressure and piston velocity with time of both engines are compared in Fig. 4. Despite the peak pressures achieved in the FPE and CSE being nearly the same, the peak pressure of the CSE achieved earlier than that of the FPE due to an earlier combustion start timing. For the CSE, the peak piston velocity during

---

**Fig. 1. FPE configuration [31].** (① Exhaust poppet valves; ② Scavenging ports; ③ Common rail fuel injection; ④ Linear electric machine; ⑤ Bounce chamber; ⑥ Bounce chamber pressure control valves; ⑦ Turbocharger compressors; ⑧ Turbocharger turbine.)

**Table 2**

| Parameters [unit]                  | Value  |
|-----------------------------------|--------|
| Maximum stroke [mm]               | 150.0  |
| Combustion chamber bore [mm]      | 131.0  |
| Scavenging ports height [mm]      | 22.0   |
| Piston and mover moving mass [kg] | 22.0   |
| Bounce chamber bore [mm]          | 150.0  |
| Exhaust back pressure [bar]       | 1.5    |
| Boost pressure [bar]              | 1.68   |

**Fig. 2. Comparison of piston velocity vs displacement.**

**Fig. 3. Comparison of cylinder pressure vs displacement.**

**Fig. 4. Cylinder pressure and piston velocity vs time.**
Table 3

Main friction mechanisms of FPE and CSE.

| Friction mechanisms       | FPE          | CSE          |
|---------------------------|--------------|--------------|
| Piston assembly           | Piston rings | Yes          |
| Crankshaft bearings       | No           | Yes          |
| Valve trains              | Yes          | Yes          |
| Linear electric generator | Yes          | No           |

The compression stroke is the same as that of the expansion stroke, while for the FPE, the peak piston velocity during the expansion process is higher than that during the compression stroke, and since the piston motion is not mechanically restricted but controlled by the instantaneous pressures acting on the piston assembly. All of the differences above are expected to contribute to differences in friction between the CSE and the FPE.

2.3. Friction mechanisms identification

A comparison of the main friction mechanisms of the FPE and CSE is shown in Table 3. The main friction mechanism for a CSE are piston assembly, valve train system and crankshaft bearings. The friction mechanism from the assembly can be divided into the top ring, second ring, oil ring (for diesel engine), and piston skirt. For the FPE, piston skirt is not necessarily to be used to guide the piston motion as the piston movement in a FPE is linear without any rotation around the piston pin, and even if a piston with a skirt is used, it will not contribute to frictional losses as there is no side forces from the connecting rod acting on the piston in the FPE. For the FPE, however, there would be friction from the linear electric generator being an integral part of the piston assembly. Numerical model for each friction mechanism and simulation results will be provided in the following sections.

3. Piston assembly friction

3.1. Methodology

3.1.1. Ring friction characteristics

There have been many theoretical and empirical models developed on piston ring assembly friction. The numerical model adopted in this research is based on fundamental lubrication theory, which has been verified by test data from a single cylinder diesel engine [40,41]. The piston ring assembly friction mainly includes friction from the piston rings and the piston skirt, which will be modelled separately. It has been reported that the lubrication between piston rings and the liner changes from the boundary condition, to mixed lubrication, then to hydrodynamic in the middle region of the piston stroke. Boundary conditions usually happen around the piston dead centres, and it is followed by mixed lubrication [40]. Hydrodynamic lubrication dominates in the full stroke [42]. An illustration of the Stribeck diagram is shown in Fig. 5.

A correlation between the friction coefficient \( f \) and the duty parameter \( S \) is developed according to the Stribeck diagram.

The duty parameter \( S \) is defined by [40]:

\[
S = \frac{\mu v_p l}{P L}
\]

(4)

where \( \mu \) is the viscosity of the lubrication oil (Unit: Pa/s); \( v_p \) is the piston velocity (m/s); \( P \) is the effective pressure behind the ring (including pressure due to ring tension and gas pressure) (Pa); and \( L \) is the height of the piston ring (m).

An SAE 30 oil is chosen as the lubrication oil due to its strong increase in viscosity with decreasing temperature. The temperature dependence of the oil viscosity at low shear rate is given by [44]:

\[
\mu (cSt) = \kappa \cdot \exp \left( \frac{\theta_1}{\theta_2 + T} \right)
\]

(5)

and

\[
\mu (\text{Pa} \cdot \text{s}) = \rho \cdot \mu (\text{cSt}) \times 10^{-6} = \rho \cdot \kappa \cdot \exp \left( \frac{\theta_1}{\theta_2 + T} \right) \times 10^{-6}
\]

(6)

where \( \kappa \) (cSt), \( \theta_1 \) (°C), and \( \theta_2 \) (°C) are parameters in the Vogel equation with a value of 0.0246, 1432.3, and 132.9 respectively; \( T \) is the liner temperature (°C); and \( \rho \) is the density of the oil, which is set to 888.0 kg/m³.

In the hydrodynamic lubrication regime, the friction force from the shearing of the viscous lubricant between the surfaces is inversely proportional to the lubrication oil film thickness, which is given by [43]:

\[
F_p = \frac{\mu A_c v_p}{h}
\]

(7)

where \( A_c \) is the lubricated contact area, and \( h \) is the oil film thickness.

Then the frictional coefficient can be written by:

\[
f_r = \frac{\mu v_p}{h P}
\]

(8)

Then

\[
h = \frac{\mu v_p}{F_p L}
\]

(9)

where

\[
f_r = C \cdot S^m
\]

(10)

\[
S = \frac{\mu v_p}{P L}
\]

(11)

\[
\mu = \rho \cdot \kappa \cdot \exp \left( \frac{\theta_1}{\theta_2 + T} \right) \times 10^{-6}
\]

(12)

3.1.2. Piston ring friction

For the hydrodynamic lubrication, \( f_r \) is almost in a linear relationship with the square root of \( S \), which can be described by [41]:

\[
f_r = C \cdot S^m
\]

(13)

where \( C \) and \( m \) are functions only of the curvature of the piston profile. In this research, \( C \) is set to 2.0, and \( m \) is set to 0.5.

A critical value of \( S \) is estimated at \( S_{cr} \approx 1.0 \times 10^{-4} \) to describe the transition from mixed to hydrodynamic lubrication. For the mixed lubrication, a linear correlation is adopted [41]:
\[ f_r = f_d (1 - S_{r0}) + f_c (S_{r0}). \]

(14)

where \( f_d \) is the dry friction coefficient of the two rubbing metal and \( f_c \) is the critical value of the friction coefficient.

In summary, the friction coefficient \( f \) can be written as:

\[
f_r = \begin{cases} f_d (1 - S_{r0}) + f_c (S_{r0}); & S < S_{\alpha} \\ C_S S_{\alpha}; & S \geq S_{\alpha} \end{cases}
\]

(15)

The final relation for the calculation of the piston ring friction force \( F_p \) is:

\[ F_p = f_r F_N (-\text{sign}(v_p)), \]

(16)

where \( F_N \) is pressure force and the tension acting on the piston ring. During the calculation of the friction force of first ring, section ring, and the oil ring, the values of \( L \) and \( F_N \) will be different.

3.1.3. Piston skirt friction

The lubrication of the piston skirt is assumed to be hydrodynamic lubrication throughout the full stroke. The friction coefficient of the piston skirt \( f_s \) is [41]:

\[ f_s = C_s \frac{\mu v_p}{P_s L_s}, \]

(17)

where \( C_s \) is the coefficient and is set to 2.5 in this research; \( P_s \) is the pressure acting on the piston skirt (Pa); and \( L_s \) is the length of the piston skirt (m).

Then the equation for the calculation of the piston skirt friction force is:

\[ F_p = f_s F_{s0} \frac{\mu v_p}{P_s L_s}, \]

(18)

where \( F_{s0} \) is the thrust force between piston and the liner (N).

3.2. Simulation results

3.2.1. Top ring friction characteristics

The friction force and the oil film thickness were calculated in Matlab with the simulation results of the FPEG operation characteristics (shown in Figs. 2–4) as input parameters. The simulation is performance with an input of piston ring height of 3.0 mm, and the simulation results for the top ring are listed in Figs. 6–8. It is found that the changing trend of the oil film is similar with that of the piston velocity. The oil film is thin around the piston dead centres, and thick around the piston middle stroke. The peak value of the oil film thickness of the CSE is higher than that of the FPE, and it is achieved during the compression stroke as well as the expansion stroke. For the FPE, the peak value reached of the oil thickness during the expansion stroke is slightly higher than that achieved during the compression stroke.

A comparison of the top piston ring friction is shown in Fig. 7. The piston ring friction force is found to reverse its direction after piston top dead centre (TDC), and reaches its peak value afterwards. The peak value achieved during the compression stroke of the FPE (350 N) is slightly higher than that of the CSE (300 N), while the peak value achieved during the expansion stroke is almost the same for these two engine types (approximately 450 N). The friction force around the piston bottom dead centre (BDC) is minimal compared with the peak friction force achieved.

The frictional power loss of the top piston ring for the CSE and FPE is compared in Fig. 8. The frictional power is calculated by the instantaneous piston velocity and the friction force of the top piston ring. It is found that the peak frictional power of the FPE (around 2550 W) is much higher than that of the CSE (approximately 1750 W). As the peak friction force of the two engines is the almost the same from Fig. 7, the reason for the frictional power difference is due to the velocity difference. As has been shown in Fig. 2, the piston velocity of the FPE is much higher than that of the CSE shortly after piston TDC, i.e. when the peak friction force is achieved.

The frictional power loss of the top piston ring for the CSE and FPE is compared in Fig. 8. The frictional power is calculated by the instantaneous piston velocity and the friction force of the top piston ring. It is found that the peak frictional power of the FPE (around 2550 W) is much higher than that of the CSE (approximately 1750 W). As the peak friction force of the two engines is the almost the same from Fig. 7, the reason for the frictional power difference is due to the velocity difference. As has been shown in Fig. 2, the piston velocity of the FPE is much higher than that of the CSE shortly after piston TDC, i.e. when the peak friction force is achieved.

The frictional power loss of the top piston ring for the CSE and FPE is compared in Fig. 8. The frictional power is calculated by the instantaneous piston velocity and the friction force of the top piston ring. It is found that the peak frictional power of the FPE (around 2550 W) is much higher than that of the CSE (approximately 1750 W). As the peak friction force of the two engines is the almost the same from Fig. 7, the reason for the frictional power difference is due to the velocity difference. As has been shown in Fig. 2, the piston velocity of the FPE is much higher than that of the CSE shortly after piston TDC, i.e. when the peak friction force is achieved.

The frictional power loss of the top piston ring for the CSE and FPE is compared in Fig. 8. The frictional power is calculated by the instantaneous piston velocity and the friction force of the top piston ring. It is found that the peak frictional power of the FPE (around 2550 W) is much higher than that of the CSE (approximately 1750 W). As the peak friction force of the two engines is the almost the same from Fig. 7, the reason for the frictional power difference is due to the velocity difference. As has been shown in Fig. 2, the piston velocity of the FPE is much higher than that of the CSE shortly after piston TDC, i.e. when the peak friction force is achieved.
that of the cylinder pressure. Piston ring friction force is minimal during the hydrodynamic lubrication process despite with a high piston velocity, while a high cylinder pressure will reduce the duty parameter, which will bring the lubrication condition to mixed lubrication from hydrodynamic lubrication. As a result, both the load acting on the piston ring and the friction coefficient will be increased, which will then increase the friction force significantly.

3.2.2. Total piston ring friction

The frictional work of one piston ring, \( W_f \) can be calculated by:

\[
W_f = \int_{t_0}^{t_f} F_f \, dx,
\]

(19)

where \( t_0 \) (s) is the time when the operation starts; \( t_f \) (s) is the time when one operation ends; and \( x \) (m) is the piston displacement.

Then the average power loss due to friction from one piston ring, \( P_f \) is given by:

\[
P_f = \frac{W_f}{T} = \frac{\int_{t_0}^{t_f} F_f \, dx}{T},
\]

(20)

where \( T \) (s) is the operation duration.

From the literature, a typical ring friction contribution is: top ring 13%, second ring 12%, and oil ring 75% [45]. Due to the limitation of the piston ring parameters, the friction force and the friction loss for the second ring and oil ring will be estimated using the simulation results of the top ring as well as the proportion above. The frictional losses from each piston ring for the CSE and FPE are compared in Table 4. It is found that the FPE doesn’t show any advantage on piston ring friction force over the CSE, and the frictional loss from the piston ring is even higher. Also, the cylinder pressure makes significant contribution to the piston ring friction force for the FPE, in conjunction with the piston velocity. As a result, it is not accurate to model the friction force to be linear to the piston velocity only, as is common in the modelling of FPEs.

3.2.3. Piston skirt friction

The friction force from the piston skirt of the CSE is shown in Fig. 11, and the frictional power loss from the piston skirt is shown in Fig. 12 with a piston skirt length of 120 mm. As the lubrication between the piston skirt and the liner is assumed to be hydrodynamic lubrication, the peak friction force from the piston skirt is lower than that from the piston ring.

For traditional reciprocating engines, the function of piston skirt is to guide the piston motion in liner and balance the lateral force caused by the crank connecting rod mechanism, while the connecting rod is attached to the piston by the piston pin, and the piston can rotate

| Parameter [Unit] | CSE   | FPE   |
|------------------|-------|-------|
| Top ring frictional loss [W] | 298.9 | 323.6 |
| Second ring frictional loss [W] | 275.9 | 298.7 |
| Oil ring frictional loss [W] | 1724.6 | 1866.7 |
| Total ring frictional loss [W] | 2299.4 | 2489.0 |

Table 4

Frictional loss from piston rings.
around the piston pin. However, as there is no side forces act on the piston in the engine of FPE and the movement of the piston is linear, the piston skirt could be removed, and the corresponding friction from the piston skirt could be eliminated. As a result, the frictional loss from the piston skirt calculated using Eq. (18) is 810.2 W for the CSE, and 0 W for the FPE.

4. Valve train friction

The correlation used to estimate the valve train friction is shown in the equation below (which excludes the camshaft bearing losses), and it is about two-thirds of the total valve train frictional losses:

\[
fmep_v = \frac{C_v \left[1 - 0.133(N/1000) n_s D_v^{0.75}\right]}{B^2 L}
\]

where \( fmep_v \) is the mean effective pressure due to valve train frictional loss (kilopascals), \( C_v \) is a coefficient \((1.2 \times 10^8\) with \( fmep_v \) in kilopascals); \( N \) is the engine speed (revolutions per minute); \( n_s \) is the number of valves; \( D_v \) is the valve diameter (mm); \( B \) is the engine bore (mm); \( L \) is the engine stroke (mm).

Then the total frictional losses from the valve train system (include the camshaft bearing), \( P_f \) can be predicted by:

\[
P_f = \frac{3}{2} fmep_v (\text{kPa}) V_e (\text{dm}^3) N (\text{rev/s})
\]

where \( V_e \) is the working volume of the engine.

The engine size and the valve configuration of the CSE are assumed to be the same with that of the FPE, then frictional losses from the valve train system of the CSE is supposed to be equal with FPE, which is estimated to be 1724.4 W.

5. Crank and bearing friction for CSE

The crankshaft friction contributions in the CSE come from journal bearings, i.e. the connecting rod, the main and accessory or balance shaft bearings, and their associated components. The journal bearings are usually designed to provide minimum film thickness of approximately 2.0 \( \mu \)m [27]. The friction force in the bearing is given by [27]:

\[
F_b \approx \left( \frac{\pi D_b L_b \mu}{h} \right) = \frac{\pi \mu D_b^3 L_b N}{h},
\]

where \( D_b \) and \( L_b \) are the bearing diameter and length, \( h \) is the mean radial clearance, and \( N \) is the shaft rotational speed.

On this basis, the frictional loss from the crank and bearing mechanism is estimated to 1508.9 W for the CSE.

6. Linear electric generator friction for FPE

From the literature, there have been several types of linear electric machine proposed for the FPE, and it was found that the appropriate machine for this device that could meet the requirement was the permanent magnetic machine [33]. Moreover, a tubular cross section of the translator was suggested due to high forces during the combustion process. The forces acting on the translator would be equally distributed, which would have minimum mechanical impact on the translator. Meantime, the net radial force between the armature and stator will be eliminated, and the friction force between them will be minimal compared with the ring friction force.

As can be found in the literature, the engine cold start-up of the FPE could be implemented by operating the linear electric machine as a motor to drive the piston reciprocate, and then reach the condition for ignition. Successful engine could start-up has been reported with a constant motor force of 110 N to overcome the ring friction force as well as the compression force of the gas [8]. Meanwhile, the piston with mover can be driven to move with a constant motor force of 60 N [2,33]. As a result, the friction force from the linear electric generator is supposed to be lower than 60 N, and it is assumed to be 30 N during the operation of the system. The frictional loss from the linear electric generator will be 265.4 W from Eq. (18).

7. Discussions and conclusion

In this research, the friction mechanisms in the FPE and CSE are investigated and compared. The main friction mechanisms are identified to be the piston assembly including three piston rings and piston skirt, the valve train system, the crank and bearing system for the CSE, and the linear electric generator for the FPE. Simulation models for each friction mechanism are discussed, and frictional losses are calculated and compared in Table 5. It is noted that the engine indicated power of the FPE is somewhat higher than that of the CSE, which can also be found from the pressure–displacement diagram shown in Fig. 3 that the area enclosed of the FPE is larger. The FPE doesn’t show advantage on piston ring friction force over the CSE, and the frictional loss from the piston ring is even higher. However, the elimination of the crankshaft system reduces the frictional loss of the FPE, and the total frictional loss of the FPE is nearly half of the CSE.

Acknowledgements

This work was funded using the EPSRC (Engineering and Physical Sciences Research Council) Impact Acceleration Account EP/K503885/1. Data supporting this publication are openly available under an Open Data Commons Open Database License. http://dx.doi.org/10.17634/154300-85. Please contact Newcastle Research Data Service at rdm@ncl.ac.uk for access instructions.

Appendix A. Supplementary material

Supplementary data associated with this article can be found, in the online version, at http://dx.doi.org/10.1016/j.applthermaleng.2018.05.018.

Table 5

| Frictional loss comparison. | CSE | FPE |
|----------------------------|-----|-----|
| Piston rings frictional loss [W] | 2299.4 | 2489.0 |
| Piston skirt frictional loss [W] | 810.2 | 0 |
| Crank and bearing frictional loss [W] | 1508.9 | 0.0 |
| Valve train system [W] | 1724.4 | 1724.4 |
| Linear electric generator frictional loss [W] | 0.0 | 265.4 |
| Total frictional loss [W] | 6342.9 | 4748.8 |
| Engine indicated power [W] | 44070.0 | 50064.0 |
| % of total frictional loss to indicated power [%] | 14.4 | 8.9 |

References

[1] Zhenfeng Zhao, Fujun Zhang, Ying Huang, Changlu Zhao, Determination of TDC in a hydraulic free-piston engine by a novel approach, Appl. Therm. Eng. 70 (1) (2014) 524–530.
[2] Boro Jia, Zhengxing Zuo, Huilua Feng, Guohong Tian, Andrew Smallbone, A.P. Roskilly, Effect of closed-loop controlled resonance based mechanism to start free piston engine generator: Simulation and test results, Appl. Energy 164 (2016) 532–539.
[3] M. Raazi Hanipah, R. Mikalsen, A.P. Roskilly, Recent commercial free-piston engine developments for automotive applications, Appl. Therm. Eng. 75 (2015) 493–503.
[4] Nigel Clark, Subhash Nedkumar, Parviz Famouri, Fundamental Analysis of a Linear Two-Cylinder Internal Combustion Engine, SAE Technical Paper, 1998.
[5] Sengo Tikkanen, Matti Välenius, Hydraulic free piston engine: the power unit of the future?, in: Proceedings of the JFPS International Symposium on Fluid Power, The Japan Fluid Power System Society, 1999.
[6] Rikard Mikalsen, A.P. Roskilly, A review of free-piston engine history and applications, Appl. Therm. Eng. 27 (14) (2007) 2339–2352.
[7] Hans Thomas Aichlmayr, Design Considerations, Modeling, and Analysis of Micro-Homogeneous Charge Compression Ignition Combustion Free-Piston Engines, University of Minnesota, 2002.
