Dynamic and thermodynamic characteristics of a linear Joule engine generator with different operating conditions

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\textbf{A B S T R A C T}

The Linear Joule Engine Generator is an energy conversion system made up from a linear expander, a linear compressor, and a linear alternator. It is adaptable to variable renewable energy fuel sources, e.g. biogases, biofuels, hydrogen and ammonia, etc. In this paper, an investigation on the system dynamics and thermodynamic characteristics under different operating conditions is presented. Real time adjustable parameters were identified, i.e. the system pressure, the valve timings, and the coefficient of electric resistance force. Their influence on the indicated power of the expander, the electric power output from the linear alternator and the energy conversion efficiency are scrutinised using a validated numerical model. In order to achieve stable operation of the system, each parameter is controlled within a practical range, and optimised to maximise the electricity generation efficiency. The system pressure was proved to be the most effective parameter to alter the system power output. The indicated power of the expander with the existing dimensions can reach up to 11.0 kW by adjusting the system pressure, and it cannot exceed 8.0 kW by just tuning valve timings or optimising the coefficient of electric resistance force. The coefficient of electric resistance force is found to be the most influential parameter to maximise the electricity generation efficiency up to 80%.

\textbf{1. Introduction}

The Linear Joule Engine Generator (LJEG) concept was first proposed by the authors’ group [1], and a working prototype has been developed aiming for potential applications of micro-scale Combined Heat and Power (CHP) systems [2]. The LJEG system is considered to combine the technologies of the Joule Engine and Linear Engine Generator [3]. The Joule cycle (or Brayton Cycle) is widely employed in gas turbines, whereas a reciprocating Joule Engine uses a separate compressor and expander to improve system efficiency [3]. The intake air is compressed, and combustion takes place under constant pressure [4]. The exhaust gas drives the expander which is connected with a generator that generate electricity [5]. The Linear Engine Generator integrates a free-piston engine with a linear electric alternator, and in principle has the advantages of a compact structure, high thermal efficiency, and multi fuel/multi combustion mode potential [6].

There have been researches reported on the Linear Engine. A four-stroke linear engine generator prototype was developed with a reported generating efficiency of 32% [7]. A two-stroke dual-piston dual-cylinder type prototype was developed by Zuo et al., with successful engine cold-start-up and combustion reported [8]. A single-piston linear engine generator with gas spring prototype was developed by researcher from Toyota Central R&D Labs Inc [9]. A dual-piston with shared-combustion chamber linear engine generator was designed and tested at Sandia National Lab [10]. Linear engine combustion with homogeneous charge compression ignition model was reported with a detailed numerical approach [11]. Piston dead center determination for linear engine was investigated and new approach was proposed by Zhao et al. [12]. The LJEG system is considered to offer the advantages of both Joule Engine and Linear Engine Generator. Thus it is of potential high efficiency and is adaptable to renewable energy applications due to the external combustion process, and efficient in generating electricity [13]. To date, there has been research on the modelling and estimation on the system performance, these have been based on simple calculations without any model validation based on experimental data.

Moss et al. developed an engine preliminary design tool in Matlab for a conventional Joule Engine [3]. The code took the following input parameters, power output, engine speed, pressure ratio, cylinder stroke/bore ratios, heat-exchanger effectiveness, combustor pressure drop and peak temperature. The code was able to predict the sizes of
cylinders required at given speed and stroke/bore ratio, mechanical efficiency, valve pressure drops, size of heat exchanger, and engine efficiency. The designed Joule Engine was found to be well suited to small CHP systems with a potential electrical generation efficiency of 33%. Multi-fuel capability was expected to burn natural gas, bio fuels, hydrogen, etc. The system efficiency was found to depend on the thermodynamic parameters such as pressure ratio, effectiveness, peak temperature. It was also affected by the system mechanical losses, which could be reduced with low mean effective pressure. The cylinder clearance volumes were suggested to be kept as small as possible to improve the volumetric efficiency and minimise the required cylinder size [3].

A conventional Joule Engine was also designed by researcher at Plymouth University [4]. A mathematical model of the system was developed, aiming to simulate the actual process and predict the performance. The modelling was undertaken in three stages: a basic engine model was used to determine the likely performance, then a more detailed model was adopted to calculate the specific losses and thus refine the basic engine model, the detailed engine model was finally used to determine the performance of the CHP system. The basic engine model considered the ideal p-V diagram to determine the ideal work per cycle. The detailed model took into account the frictional, thermal, and pressure losses, and it predicted that the engine was suitable for micro CHP applications with a maximum thermal efficiency of 33%. The engine mathematical model was validated through the processes of design, build, and testing of both an engine for technology demonstration and a prototype engine. When the sub-models were combined to model a CHP system, with the engine exhaust heat used to preheat the combustion air, the overall maximum system efficiency achiever could be up to 79% [14].

The Linear Joule Engine Generator concept was first proposed by the authors’ group, initially aiming for application for micro-scale CHP generation [1]. Simple calculations were undertaken, and the simulation results suggested that a domestic CHP plant based on the proposed technology could reach an electric generating efficiency of above 30%. With a heating temperature of around 1100 K and a compressor outlet pressure of 6 bar, the engine was able to produce 4.5 kW of mechanical power. Whilst, through waste heat recovery technology, the total system could show a promising efficiency of over 90%. Later on, a 3-dimensional diagram of the proposed LJEG system was presented by the authors [2]. The geometric parameters of the system were optimised in LMS AMESim software, which provide a solid basis for the manufacturing of the prototype. Meanwhile, Wu et al. presented a coupled dynamic model of the Linear Joule Engine and the connected permanent magnet linear electric generator, aiming to provide a better prediction of the system performance. It was estimated that the LJEG system could generate 1.8 kW electricity, with an engine thermal efficiency of 34% and electric generating efficiency of 30% [15].
The reported studies on this area were mainly based on simple calculations with fundamental thermodynamic equations, very few of them was validated against experimental data. In this paper, detailed dynamic and thermodynamic characteristics of a LJEG system with different operating conditions is presented. The adjustable operating parameters are identified, and their effects on the system power output and efficiency are evaluated based on a validated numerical model that has been published in a previous paper [16]. A sensitivity analysis of parameters is conducted, aiming to provide a guidance on the performance under various working conditions and the control strategy optimisation of the LJEG system.

2. System configuration and model description

In this section, the LJEG prototype developed at Newcastle University will be presented. A 0/1 dimensional numerical model is developed in Matlab/SIMULINK, the fundamental equations used to describe the system dynamic and thermodynamic processes are summarised.

2.1. System configuration

The LJEG prototype developed by the authors is shown in Fig. 1, with the system specifications shown in Table 1. Two double-acting free-pistons are placed in the compressor (left) and the expander (right) respectively, which separate the cylinders into two opposite chambers. The thermodynamic cycle starts from the compressor, where air is compressed and fed to the external reactor to react with fuel. The high pressure, high temperature gas from the reactor goes into the expander, and drives the linear alternator to generate electricity and power the compressor. Detailed information of the system configuration and the numerical model with further validation can be found in the literature [16].

2.2. Model description

The numerical model aims to describe the dynamic and thermodynamic characteristics of the LJEG system, e.g. the piston motion, the pressure variation in the expander and the compressor, the power output, the system efficiency, etc. As the piston in the proposed system is not restricted by a mechanical linkage, the piston motion is determined by the forces acting on it, which are the gas pressure forces from the linear expander and the compressor, the resistance force from the linear alternator, the frictional force, and the inertia of the moving mass, which can be expressed as below according to Newton’s Second Law:

\[ F_{\text{exp}} + F_{\text{com}} + F_{p} + F_{f} = m\ddot{x} \]  
\[ F_{\text{exp}} = F_{\text{exp},l} + F_{\text{exp},r} \]  
\[ F_{\text{com}} = F_{\text{com},l} + F_{\text{com},r} \]  

where \( F_{\text{exp}} \) (N) is the pressure force from the linear expander; \( F_{\text{exp},l} \) (N) is the pressure force from the left chamber of the linear expander; \( F_{\text{exp},r} \) (N) is the pressure force from the right chamber of the linear expander; \( F_{\text{com}} \) (N) is the pressure force from the linear compressor; \( F_{\text{com},l} \) (N) is the pressure force from the left chamber of the linear compressor; \( F_{\text{com},r} \) (N) is the pressure force from the right chamber of the compressor. Detailed numerical models to describe cylinder pressure force from expander and compressor, friction force, and resistance force from the linear alternator can be found in the previous publications.

The gas forces from both chambers of the linear expander and compressor can be calculated by the gas pressure and piston effective area, where can be represented as following:

\[ F_{\text{exp},l} = p_{\text{exp},l} A_{\text{exp}} \]  
\[ F_{\text{exp},r} = p_{\text{exp},r} A_{\text{exp}} \]  
\[ F_{\text{com},l} = p_{\text{com},l} A_{\text{com}} \]  
\[ F_{\text{com},r} = p_{\text{com},r} A_{\text{com}} \]  

where \( p_{\text{exp},l} \) (Pa) is the cylinder pressure from the left chamber of the linear expander; \( p_{\text{exp},r} \) (Pa) is the cylinder pressure from the right chamber of the linear expander; \( p_{\text{com},l} \) (Pa) is the cylinder pressure from the left chamber of the linear compressor; \( p_{\text{com},r} \) (Pa) is the cylinder pressure from the right chamber of the linear compressor; \( A_{\text{exp}} \) (m³) is the piston area of the expander; \( A_{\text{com}} \) (m³) is the piston area of the compressor.

By applying the first law of thermodynamics on the charge in the chamber and ideal gas equation, yields the pressure calculation equation for one of the two chambers of the linear expander [16]:

\[ \frac{dp_{\text{exp}}}{dt} = \frac{\gamma - 1}{V_{\text{exp}}} \left( \frac{dQ_{\text{hi}}}{dt} \right) \frac{p_{\text{exp}} \gamma dV_{\text{exp}}}{V_{\text{exp}} \gamma} + \frac{\gamma - 1}{V_{\text{exp}}} \sum_{i} m_{\text{eqt}} h_{\text{eqt}} \]  

where \( p_{\text{exp}} \) is the pressure in the chamber of the linear expander (pa); \( \gamma \) is the heat capacity ratio; \( V_{\text{exp}} \) is the working volume of the linear expander for one cylinder (m³); \( Q_{\text{hi}} \) is the heat transfer between the internal gas and the cylinder (J); \( m_{\text{eqt}} \) is the mass flow rate in or out of the valve (m/s); \( h_{\text{eqt}} \) is the specific enthalpy of the mass flow (kJ·kg⁻¹).

For the linear compressor, the relationship between gas pressure \( p_{\text{com}} \) and volume of the chamber \( V_{\text{com}} \) during the compression/expansion process is listed below [16]:

\[ \frac{dp_{\text{com}}}{dt} = \frac{\gamma - 1}{V_{\text{com}}} \left( \frac{dQ_{\text{hi}}}{dt} \right) \frac{p_{\text{com}} \gamma dV_{\text{com}}}{V_{\text{com}} \gamma} \]  

where \( p_{\text{com}} \) is the pressure in the chamber of the linear compressor (pa); \( \gamma \) is the heat capacity ratio; \( V_{\text{com}} \) is the working volume of the linear compressor for one cylinder (m³).

The total friction force \( F_{f} \) of each piston is estimated as linear combination of piston velocity plus a constant\( C_{f} \), as shown in the equation below [17]:

\[ F_{f} = -\left(C_{k} \frac{dv}{dt} + C_{s}\right) \text{sign}(v) \]  

\( C_{k} \) is the kinetic friction coefficient related to the instantaneous velocity, and the \( C_{s} \) is the static friction coefficient as a constant part of the frictional force.

The linear electric machine is operated as a generator, electrical current is drawn from the alternator coils through the continuous back and forth movement of the mover [18]. The linear alternator is modelled using a simplified numerical model to make it feasible with limited amount of design parameters known to the users. Fig. 4 illustrates an equivalent circuit of the linear electric machine.

As the resistance force from the electric machine is assumed to be

<table>
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</tr>
<tr>
<td></td>
<td>Outlet pressure [bar]</td>
<td>7.0</td>
</tr>
</tbody>
</table>
proportional to the current of the circuit according to electromagnetic theory, the resistance force from the generator is then written as [19]:

\[ F_i = -C_e v \]  

(11)

where \( C_e \) is the coefficient of the electric resistance force from the linear alternator, which can be calculated from the physical parameters of the alternator design specifications and the external load of the circuit [20].

For the calculation of the system efficiency, as illustrated in Fig. 2, the fuel chemical power is assumed to be converted to the indicated power of the linear expander with an indicated efficiency of \( \eta_i \). Then the indicated power of the linear expander is converted to be electric power by the linear alternator with an electricity generation efficiency of \( \eta_e \). As the reactor in the designed LJEG system is adaptive to a variety of fuels and renewable sources, the system indicated efficiency is not discussed in this research as it is determined to the specific fuel/source adapted in the reactor.

The electricity generation efficiency from the expander indicated power, \( \eta_e \), is expressed as:

\[ \eta_e = \frac{P_e}{P_{ex}} \]  

(12)

The simulation model is developed in Matlab/Simulink. Both the piston displacement and velocity generated in the simulation are monitored and fed back to a controller which determines the valve timings. The initial piston position is assumed to be at its top dead centre (TDC), and the initial clearance from the left cylinder head in the linear expander is 8.0 mm. The expansion process of the expander is initialised after the intake valve open (IVO), which is actuated when the piston reaches its TDC. The exhaust valve open (EVO) of the left chamber of the linear expander is triggered when the piston reaches its bottom dead centre (BDC). The simulation model has been validated against test data from a conventional reciprocating Joule Engine, and a LJEG prototype developed by the authors’ group. Detailed validation results can be found in elsewhere [21].

3. Operating conditions

The operating parameters of the LJEG system can be adjusted in the operation, while the dimension parameters are fixed which cannot be changed during the operation. By analysing the components of the system and its working principle, it is found that the system performance could be influenced by the variable system operating parameters, i.e. the system pressure, the valve timing, and the coefficient of electric resistance force. The system pressure and valve timing are used to adjust the energy input, the coefficient of electric resistance force is used to vary the energy output. The influence of the variable operating parameters to the system performance will be discussed individually.

3.1. System pressure

The system performance with different system pressures is shown in Figs. 3–5, with the other operating parameters remain unchanged. The LJEG system shows good additivity to a wide range of inlet pressure levels, which is regarded as “system pressure” in this paper. The range of the system pressure is primarily set from 5.0 bar to 9.0 bar. The pressure of 9.0 bar represents that can be reasonably achieved in a single stage compressor. With the current fixed coefficient of electric resistance force, the system pressure is required to be above 5.0 to
ensure a stable operation; otherwise, the pressure force from the expander is supposed to be insufficient to drive the electric alternator and overcome the compression force of the compressor and frictional force.

From Fig. 3, it is found that, with higher system pressure, piston stroke increases. The area enclosed by the pressure-displacement curve becomes larger with higher system pressure, indicating a higher indicated work per cycle is achieved. When the system pressure is higher than 7.5 bar, the peak cylinder pressure in the expander will be higher than the system pressure at the beginning of the expansion process. This phenomenon is more obvious when the system pressure is above 9.0 bar, and the peak cylinder pressure reaches 11.0 bar at the end of the compression process of the expander after the exhaust valve closes. The high compression pressure here is to overcome the high expansion force in the expander of the other side, and slow down the piston. As a result, the coefficient of electric resistance force is suggested to be increased in this case, in order to absorb more expansion energy from the expander, and generate more electricity.

The power output with different system pressures is shown in Fig. 4, with a linear fitting curve compared in the same figure. It is observed that, both the indicated power of the expanders and the electric power of the linear electric alternator show linear increase with the system pressure. When the system pressure increases from 5.0 bar to 9.0 bar, the indicated power of the expander changes from approximately 3.0 kW to 11.0 kW, and the electric power improved from 2.0 kW to around 7.0 kW. When the system pressure is increased to above 7.5 bar, the electric power extracted from the LJEG system can be above 5.0 kW.

As a result, with the current setting of the system volumetric parameters and operating parameters, the indicated power of the linear expander, $P_e$ (W) can be estimated by:

$$P_e = 1943.8 \times P_{in} - 6848.1$$

(13)

The electric power output of the linear alternator, $P_l$ (W) can be estimated by:

$$P_l = 1247.4 \times P_{in} - 4214.9$$

(14)

where $P_{in}$ (bar) is the inlet pressure of the linear expander, or the system pressure.

The electricity generation efficiency and the frequency with different system pressures are demonstrated in Fig. 5. The system frequency increases with the system pressure and shows a linear relationship, while the efficiency drops gradually with a higher system frequency. As a result, in order to achieve a higher electricity generation efficiency, the coefficient of electric resistance force is suggested to be increased gradually with the system pressure.

### 3.2. Valve timing

The intake valve duration of the expander is illustrated in Fig. 6, the piston displacement and velocity are used as feedbacks for the valve timing control. It is open when the piston reaches its top dead centre, and the piston velocity reduces to zero. The valve response time is not considered in this research, and the valves are assumed to be completely open/close according to the open/close command in the simulation model. Eight different cases with different intake valve closing timings are investigated, with the intake valve closing position summarised in Table 2. The intake valve closing position in Table 2 is illustrated with its relative distance from the mid-point of the stroke (marked as central stroke in Fig. 6). From Case 1 to Case 8, the intake valve is control to close earlier, and the intake valve duration is reduced gradually. The system pressure is set to 7.0 bar and the other parameters remain unchanged during the simulation.

The cylinder pressure in the expander with different intake valve closing positions is shown in Fig. 7. The simulation results for case 1 and case 8 will be discussed later separately. With a later intake valve closing timing, the piston stroke will be longer and the indicated work generated from the expander will be higher. Meanwhile, the cylinder pressure at TDC before the intake valve opens, as well as the cylinder pressure at BDC before the exhaust valve closes will be higher with a later intake valve closing position (the intake valve closes from −5 mm to 20 mm away from the mid-point of stroke). The relationship of the system power output and efficiency with different valve timings is shown in Fig. 8. It is found that the efficiency decreases with longer intake valve duration. It is suggested that a larger electric output alternator with a higher coefficient of electric resistance force will be coupled with the Linear Joule Engine to extract more energy from the

### Table 2

<table>
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<th>Case number</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
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<th>6</th>
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<td>10</td>
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<td>−5</td>
<td>−10</td>
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Fig. 6. Illustration of expander intake valve timing.

Fig. 7. Expander cylinder pressure with different valve timings.

Fig. 8. Power output with different valve timing.
expander. With an over long intake valve duration, as Case 1 in Table 2, the cylinder pressure in the expander will be higher than the system pressure as shown in Figs. 9 and 10, which is marked as “Over compression” in the figures. As a comparison, the simulation result for Case 2 is drawn in the same figure. When the energy input/output balance is changed, whether the input energy is increased or the output energy is decreased, “over compression” will take place to increase the compression energy to compensate the energy balance and contribute the net force to reduce the piston velocity to 0. When “Over compression” happens, the piston fails to reverse its direction before the expander gas pressure reaches the intake pressure level, the gas in the expander cylinder will flow to the intake pipe when the intake valve opens, and the expander cylinder gas decreases when its working volume increases. As a result, the combustion gas cannot enter the expander until the expander pressure drops below the intake pressure. If “Over compression” takes place, the system energy input is suggested to be reduced or the system energy output is suggested to be increased by adjusting the corresponding operating parameters as discussed in Section 2.3.

If the intake duration is overly short, as Case 8 in Table 2, the piston dynamics is shown in Fig. 11 with the results for Case 4 (stable operation) compared in the same figure. When the intake valve duration is too short, excitation force from the expander is insufficient to maintain stable operation of the system, then piston stops after one operation cycle. As a result, the intake valve of the expander is suggested to open with reasonable duration if the other operating parameters remain the same in order to ensure the system stable operation and to optimise electricity generation efficiency.

3.3. Electric resistance force

The system performance with different coefficients of electric resistance force is simulated, and the other input parameters remain unchanged during the simulation. The system pressure is set to 7.0 bar, and the valve timing is set as Case 4 in Table 2. The values for different coefficients of electric resistance force are summarised in Table 3, where $\alpha$ is scale factor of the coefficient of the electric resistance force to its maximum value, $C_{\text{max}}$, which can be expressed as:

$$C_r = \alpha \cdot C_{\text{max}}$$  \hspace{1cm} (15)

The piston dynamics with different coefficients of electric resistance force are shown in Fig. 12. It is found that the both the piston amplitude and the peak piston velocity are higher with a lower coefficient of electric resistance force. The peak piston velocity is achieved after the piston crosses the middle stroke. The piston normally has a prompt deceleration than its acceleration due to intake valve timing.

The system power output with different coefficients of electric resistance force is shown in Fig. 13. When coefficient of electric resistance force is increased, the expander indicated power drops, while the electric power generated from the linear alternator increases. The electricity generation efficiency shows a linear increase with the coefficient of electric resistance force. While the coefficient of electric resistance force is increased to the value of 816.9 in the Case 12 in Table 3, piston stops after one operation cycle due to the high resistance force from the linear alternator (Fig. 14). As a result, the coefficient of electric resistance force is suggested to be limited to a proper range in order to avoid unstable operation, and within this proper range, it is suggested to be maximised to improve the electricity generation efficiency.

4. Discussions

As discussed in the above section, variations on the system pressure, valve timing, and coefficient of electric resistance force will affect the system performance. In order to ensure stable operation of the system, each parameter is supposed to be controlled within a reasonable range, and within this particular range, the influence of each variable parameter on the system performance is different. The system power output with different adjustable parameters is simulated and compared, aiming to explore the sensitivity of each variables. The changing range of system pressure is 5.0–9.0 bar; the changing range of intake valve closing position is 25 to −5 mm away from midpoint of stroke; the
The comparison on power output is shown in Fig. 15. It is found that the system pressure is the most effective parameter to achieve a higher indicated power from the expander and a thus a higher electric power from the linear alternator. By improving the system pressure, the expander indicated power can reach up to 11.0 kW with the other parameters remain unchanged; while the expander indicated power cannot exceed 8.0 kW by varying valve timing or coefficient of electric resistance force.

A comparison on electricity generation efficiency with different operating parameters is shown in Fig. 16. The coefficient of electric resistance force is found to be the most influential parameter for the electricity generation efficiency, and an efficiency of 80% is achievable by increasing the coefficient of electric resistance force. By changing valve timing, the electricity generation efficiency can be improved to a maximum of 75%. The influence of the system pressure on the electricity generation efficiency is moderate, which varies from 65% to 70% during the operating range despite its significant effect on system power output.

5. Conclusions

In this paper, the system dynamic and thermodynamic characteristics of a LJEG system with different operating conditions is presented. The adjustable operating parameters were identified as system pressure, valve timing, and coefficient of electric resistance force. Main conclusions from this research are listed below:

(1) In order to ensure stable operation of the system, each parameter is to be controlled within a reasonable range, and within a proper range it is to be optimised to improve the electricity generation efficiency.

(2) With higher system pressure, the indicated power from the expanders and the electric power from the linear alternator increase proportionally. While over high system pressure will lead to higher peak cylinder pressure in the expander in order to slow down the piston and overcome the high expansion force in the expander of the other side. In this case, the coefficient of electric resistance force is suggested to be increased to improve the electricity generation efficiency.

(3) With a longer intake valve opening duration, piston stroke of the expander will be extended, and more indicated power and electric power will be extracted. The electricity generation efficiency can be improved with a higher coefficient of electric resistance force.

(4) It is found that the both piston amplitude and peak piston velocity are higher with a lower coefficient of electric resistance force. When the coefficient of electric resistance force is increased, the expander indicated power drops, while the electric power generated from the linear alternator increases. The electricity generation efficiency shows a linear increase with the coefficient of electric resistance force.

(5) The system pressure is the most effective parameter to improve system power output. The thermal power can reach up to 11.0 kW by improving the system pressure; while cannot exceed 8.0 kW by varying valve timing or coefficient of electric resistance force. The coefficient of electric resistance force is found to be the most significant parameter for the electricity generation efficiency and an

Table 3

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<th>Case number</th>
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<td>694.4</td>
<td>735.2</td>
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Fig. 12. Piston dynamics with different coefficients of electric resistance forces.

Fig. 13. Power output with different coefficient of electric resistance forces.

Fig. 14. Piston dynamics with over high coefficient of electric resistance force.
efficiency of 80% is achievable by increasing the coefficient of electric resistance force.

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