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INTEGRATING COMPRESSED AIR ENERGY STORAGE WITH A DIESEL ENGINE FOR ELECTRICITYGENERATION IN ISOLATED AREAS

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Abstract

This paper reports an integrated system consisting of a diesel genset and a Compressed Air Energy Storage (CAES) unit for power supply to isolated end-users in remote areas. The integration is through three parts: direct-driven piston-compression, external air turbine-driven supercharging, and flue gas waste recovery for super-heating. The performance of the integrated system is compared for a single diesel unit and a dual-diesel unit with a capacity of electricity supply to a village of 100 households in the UK setting. It is found the fuel consumption of the integrated system is only 50% of the single-diesel unit and 77% of the dual-diesel unit. The addition of the CAES unit not only provides a shift to electrical energy demand, but also produces more electricity due to the recovery of waste heat.

Key words: district energy supply; compressed air energy storage; thermal energy storage; supply side management; system integration.

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Nomenclature	
A	Constant Eq. (8) (K)
h	Specific enthalpy (J kg^{-1})
h_v	Volumetric heat transfer coefficient (W m^{-3} K^{-1})
$\dot{m}$	Mass flow rate (kg s^{-1})
m	Mass (kg)
n	Polytrophic factor
N	Number of stages (-)
P	Pressure (Pa)
R	Universal gas constant (J kg^{-1} K^{-1})

T	Temperature (K)
t	Time (s)
V	Volume (m^3)
W	Power (W)
Superscripts	
rated	Rated conditions
max	maximum
Subscripts	
a	Air
AT	Air turbine
am	Ambient conditions
ex	Exhaust
f	Fuel
DE	Diesel engine
EU	End user
IC	Inlet compressor
LHV	Lower heating value
PC	Piston compressor
PCM	Phase change material
s	storage
SD	Supercharged diesel engine
Greek letters	
γ	Adiabatic index (-)
η	Diesel engine thermal efficiency (-); isoentropic efficiency (-)
λ	Air to fuel ratio (-)
ξ	Constant parameter Eq. (2)
ρ	Density (kg m^{-3})
σ	Normalized standard deviation (-)

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1. Introduction

The use of diesel generators is a preferred option for electricity production in remote areas where the cost of national grid extension is prohibitively expensive [1-4]. While diesel power generating unit requires relatively little investment, the fuel costs increase by up to a multiple of six to ten when the associated transportation charges are taken into account [2, 5]. Therefore operating a diesel power generator at a higher efficiency is critical for saving fuel cost, which also brings environmental benefits. A typical load pattern for remote area power supplies (especially for village scales) is characterized by a small to medium base load, and several periods of high electricity demand during a day [1, 6]. In addition with the intermittent renewable electricity generation such as wind power, in most cases diesel generators have to be operated at a low load factor for most of the time. Figure 1 shows fuel consumption and efficiency characteristics of a typical diesel engine operated at different load factors (as described in details in Section 3.1). It can be seen that, for a low- and medium-penetration system, the diesel fuel consumption even at zero load, is approximately 35% of that at the rated power output. Moreover, operating a diesel generator at light loads (< 30-50% of rated load) can accelerate carbon deposits of wear and tear and thus shorten the lifetime of the equipment, leading to a high maintenance cost [7, 8]. As a consequence, interests in the integration of diesel engine with energy storage technologies have been growing enormously over the past decades. Studies have been done on enabling diesel generators to be operated above a certain minimum level of load in order to maintain an acceptable efficiency and to reduce the rate of premature failures [9-12].

Attention has also been paid to the waste heat recovery of diesel engines to enhance the overall performance. An inspection of the energy balance of internal combustion engines indicates that the input energy can be roughly divided into three equal parts: energy converted to useful work, energy transferred to coolant and energy lost through exhaust [13, 14]. Thermal energy loss from the exhaust can be regarded as a high grade, which has a temperature ranging approximately from 400 to 600°C [15]. Recent work has shown a potential increase in the overall efficiency by up to 30% through efficient recovery of waste heat [16]. Technologies proposed for the recovery of waste heat include Organic Rankine Cycle (ORC) [17], thermoelectric generation [18] and the use of heat pumps [19]. However, when a diesel engine is used for remote electricity generation, the temperature of the exhaust gas changes frequently as does the load factor [20]. Thermoelectric power generation is expensive and has a low efficiency. The unsteady exhaust gas temperature is disadvantageous for the operation of an ORC engine or a heat pump. Compressed Air Energy Storage (CAES) presents an alternative solution to the issue, which can store excessive shaft power, and recover the waste heat of the diesel engine in the energy extraction process. Using CAES to deal with the stochastic fluctuations of wind power in wind-diesel hybrid systems has been examined numerically, and the results are promising in enhancing the wind energy penetration [2, 21]. In this paper, an integrated diesel-CAES power system is proposed and investigated. The aim is to reduce fuel consumption and production costs for electricity generation in rural areas. Specific attention is paid to the operating principle and the influence of demand patterns of end-users. This may lead to a real system to be developed accordingly to demonstrate the advantages.

2. System configuration and operating principle

Most modern diesel engines are turbocharged or even supercharged. A turbocharger or a supercharger is made up of a coupled compressor-turbine unit aiming to increase the density of the engine air intake. This results in the engine producing significantly more power than a naturally aspirated engine with the same combustion-chamber volume. The difference between the turbocharger and supercharger is that the supercharger has a compressor driven mechanically by external power such as the engine's crankshaft, while a turbocharger is powered by the engine exhaust, therefore does not require any mechanical power.

This paper focuses on an integrated diesel-CAES system in which the diesel engine could be supercharged by a CAES unit, as illustrated in Figure 2. The diesel engine used in such an integrated system differs from the traditional engine in the air intake method: the atmospheric air can either be naturally aspirated or forcibly compressed into the combustion-chamber depending on the switch conditions of a 3-way valve located at the engine inlet. The CAES unit is made up of a piston compressor, a compressed air reservoir, two heat exchangers and a two-stage air turbine. The diesel engine and the CAES unit are integrated by three parts: first the diesel engine shaft and the piston compressor shaft could be mechanically connected or disconnected through using Clutch 1. Second, the air turbine shaft in the CAES unit and the compressor shaft in the diesel engine can be mechanically connected or disconnected by the use of Clutch 3. Third, the flue gas from the diesel engine and the compressed air from the reservoir are both fed into Heat exchangers 1 and 2 for waste heat recovery. It is worth mentioning that two heat exchangers are used not only for the heat transfer between the flue gas and compressed air, but also for the storage of thermal energy in cases where the engine and air turbine operate at different times. Phase Change Materials (PCMs) can be packed into the heat exchangers for high-density thermal energy storage. Examples of PCMs for such an application are composite materials consisting of an inorganic salt (PCM) and a ceramic matrix due to their favorable costs, good energy density and a wide range of melting temperatures [22].

From the above one can see there are three power-related components in the integrated system: diesel engine, piston compressor and air turbine (includes the coupled compressor for supercharging). Theoretically, based on the operating status of the three components, the system has 8 operating modes, as listed in Table 1.

Of the modes shown in Table 1, Mode 6 is the off state of the integrated system whereas Mode 7 is almost inapplicable as the air turbine is not connected to the piston compressor in the system. In addition, from an energy utilization point of view, Mode 7 is not practical as it produces nothing, but consumes compressed air, due to process irreversibility. Similarly, Mode 8 is virtually impossible as no power is available to drive the piston compressor. As a result, the integrated diesel-CAES system has the following five potentially useful operating modes:

- Mode 1 focuses on a supercharged-diesel process to respect the high demand of end-users. In this process Clutch 3 is connected while Clutch 1 and Clutch 2 are disconnected. The 3-way valve is turned towards the compressor side so that the coupled air turbine-compressor is able to intake more air to be compressed into the diesel chamber. The reason for using a coupled air turbine-compressor instead of charging the diesel chamber directly with compressed air from the reservoir, is that the air pressure in the reservoir is much higher than the required pressure of diesel chamber. With waste heat recovery from flue gas the high-pressure compressed air in the reservoir can drive the coupled air turbine-compressor to produce about 5 times of low-pressure compressed air that is required by the diesel chamber.
- Mode 2 is a mode with all the clutches disconnected and the 3-way valve turned towards the atmospheric side. As a result the diesel engine runs in a traditional manner: the atmospheric air is naturally aspirated into the cylinder for combusting the diesel

fuel. The shaft power produced by the diesel engine is used to generate electricity for end-users. The exhaust gas of the diesel engine is used to heat the PCMs in Heat Exchangers 1 and 2 in order to recover the high grade heat for later uses in the compressed air expansion process.

- Mode 3 is similar to Mode 1 but with Clutch 1 connected. The power generated by the supercharged diesel engine is used to respect the end-users' demand, as well as drive the piston compressor to produce the compressed air.
- Mode 4 is similar to Mode 2 but with Clutch 1 connected. The power generated by the diesel engine is used to respect the end-users' demand as well as drive the piston compressor to produce the compressed air.
- Mode 5 is a case with the diesel engine turned off, Clutch 1 and Clutch 3 disconnected and Clutch 2 connected. The compressed air is heated up first by the thermal energy stored in Heat exchanger 1 and Heat exchanger 2 prior to the expansion in the air turbines to drive Generator 2 to produce electricity for end-users. Such a process avoids the diesel engine to operate at a very low load factor and as a result saves on fuel consumption.

The proposed integrated diesel-CAES system is designed to match the load of typical end-users in remote areas without access to electric grids. Therefore, the operation mode of the system has to be updated regularly after each operating step. In the operational mode selection process, not only the end-user's demand, but also the pressure of the air reservoir and the status of the heat stored in the heat exchangers play decisive roles. In this study a control algorithm is developed and programmed in MATLAB. In this program, the power consumption of the piston engine, the maximum power outputs of the air turbine and the supercharged diesel engine are calculated based on the updated inputs. Table 2 presents the logic to select the operation mode at each instant of time. First pressure P_S in the CAES reservoir and end user demand W_{EU} are assessed. If P_S is higher than the maximum allowed value CAES charging is not possible; otherwise CAES charging can potentially take place in the case power output of the asset is large enough to satisfy both the end user demand and provide power to the piston compressor to charge the CAES. If this is not the case meet the satisfying the end user demand has the priority over CAES charging. When storage pressure is within the allowed range ($P_S^{\min} < P_S < P_S^{\max}$) CAES discharge can occur and if the end user demand is smaller than the power delivered by the air turbines. It should be noted that such a selection principle is based on the assumption that the rated mass flowrate of the piston engine is higher than that of the air turbine. This ensures the possibility to charge the CAES reservoir. In fact, if $\dot{m}_{PC} \neq 0$; $\dot{m}_{AT} \neq 0$ $\neq$ 0 and $\dot{m}_{PC} < \dot{m}_{AT}$ under design conditions the mass flow rate withdrawn from the CAES would be higher that the injected one without the necessary conditions for air accumulation in the reservoir.

Compared to non-charged diesel engines, the integrated diesel-CAES system can downsize the scale of the facility due to the application of supercharging, thus enabling the system to be operated at a high load factor. In addition, the integrated system could supply electric power solely by air turbine when the end-users' demand is low (Mode 5), thus avoiding the diesel engine to be operated at a very low load factor. In the following, attention is given to fuel consumption of the integrated diesel-CAES system using the results of traditional diesel sets as the baseline.

3. Thermodynamic modelling of the key processes

Numerical modeling is employed in this section to examine the effect of the use of CAES on the efficiency and fuel consumption of the diesel engine. As described in Section 2, the integration is through 3 main parts in the diesel-CAES integrated system consisting of the diesel engine, the piston-compressor and compressed air reservoir, the air turbines (including the heat exchangers) and a coupled compressor. In the following, each of the components will be numerically modeled to evaluate the overall performance of the integrated system. It should be noted that many factors influence the performance of the components including the manufacturer, the operational conditions, and sizes etc. However, as this work represents a first step towards developing such an integrated system, generic models are adopted.

3.1 Diesel engine (including the inlet compressor)

This study focuses on two main performance indicator for the diesel engine: the efficiency and the fuel consumption. In a diesel engine the fuel consumption rate is governed mechanically or electronically by a fuel injection system to meet the required load factor. In our study the diesel engine works in two different modes depending on the end-users' demand. When the demand is lower than the rated power (the maximum power output without supercharging), the supercharging unit is switched off (Mode 2 and 4). Therefore the fuel consumption and thermal efficiency of the engine could be estimated as[6, 23]:

$$\frac{\dot{m}_f}{\dot{m}_f^{rated}} = \xi + (1-\xi)\cdot\frac{W_{DE}}{W_{DE}^{\max}} \quad \text{(Eq. 1)}$$

$$\eta_{DE} = \frac{W_{DE}}{\dot{m}_f \cdot Q_{LHV}} \quad \text{(Eq. 2)}$$

In the above equations, $\dot{m}_f$ and $\dot{m}_f^{rated}$ are respectively the real-time and rated flow rate of the diesel fuel, W_{DE} and $W_{DE}^{\max}$ are respectively the real-time and rated power output, ξ is a constant related to the consumption curve of the generator and is equal to 0.34 for a non-charging diesel engine, Q_{LHV} is the lower heating value of diesel fuel equal to 43.4 MJ/kg. The rated thermal efficiency of the diesel engine $\eta^{rated} = W_{DE}^{\max} / \dot{m}_f^{rated} \cdot Q_{LHV}$ is set at 0.32 in this study. As the diesel engine operates at a constant rotational speed for electricity generation, the mass flow rate is considered to also be constant in this mode with the rated air/fuel ratio $\lambda^{rated} = \frac{\dot{m}_a^{rated}}{\dot{m}_f^{rated}}$ equal to 14.7 based on stoichiometric balance.

When the end-users' demand is higher than the rated power, the supercharging unit is switched on so more air can be blown into the combustion-chamber by the inlet compressor (operational mode 1/3). For a constant speed diesel engine with pre-cooling, the mass of air entering the engine is proportional to the inlet pressure as:

$$\frac{\dot{m}_a}{\dot{m}_a^{rated}} = \frac{P_{IC}}{P_{am}} \qquad \text{(Eq. 3)}$$

In the above equation P_{IC} is the outlet pressure of the inlet compressor, P_{am} is the ambient pressure, and $\dot{m}_a$ is the mass flow rate of air entering the engine The power consumption of the compressor is featured with isentropic efficiency. This is the comparison between the actual performance and the performance that would be achieved under idealized circumstances (isentropic processes) for the same inlet state and the same exit pressure. In order to meet the users' demand at all times, the compressor has to operate under off-design conditions using different outlet pressure and flow rate. The isentropic efficiency η_{IC} of the inlet compressor varies with the real-time outlet pressure and can be evaluated as [24]:

$$\frac{\eta_{IC}}{\eta_{IC}^{ref}} = 1 - \left(\sqrt{\frac{\Delta h_s^{ref}}{\Delta h_s}} - 1 \right)^2 \qquad \text{(Eq. 4)}$$

In the above equation Δh_s indicates the enthalpy change in an isentropic process, while the superscript ref denotes the reference state at the design point or process. The necessary work for compression then can be expressed as:

$$W_{IC} = \dot{m}_a \cdot W_s / \eta_{IC} \qquad \text{(Eq. 5)}$$

In the above equation W_s is the specific power consumption in an isentropic process and W_{IC} is the actual power required for the compression process

The thermal efficiency η_{SD} of the diesel engine relates to the real-time air/fuel ratio $\lambda = \frac{\dot{m}_a}{\dot{m}_f}$ following a quadratic model [2]:

$$\eta_{SD} = \left(0.55 - 0.23 \cdot \left[{(\lambda - 53.0)}\big/{38.3} \right]^2 \right) \qquad \text{(Eq. 6)}$$

The power generated by the engine when supercharged can then be calculated by:

$$W_{SD} = \dot{m}_f^{rated} \cdot \eta_{SD} \cdot Q_{LHV} \qquad \text{(Eq. 7)}$$

The temperature of the exhaust gases can also be calculated using the following estimate [2]:

$$T_{ex} = T_{am} + \frac{A}{1 + \lambda \cdot B} \qquad \text{(Eq. 8)}$$

In the above equation T_{am} is the ambient temperature. A and B are constants which equal to 1000K and 0.0667 based on the experimental tests.

It should be noted that such a selection principle is based on the assumption that the rated mass flowrate of the piston engine is higher than that of the air turbine. This ensures the possibility to charge the CAES reservoir. In fact, if $\dot{m}_{PC} \neq 0$; $\dot{m}_{AT} \neq 0 \neq 0$ and $\dot{m}_{PC} < \dot{m}_{AT}$ under design conditions the mass flow rate withdrawn from the CAES would be higher that the injected one without the necessary conditions for air accumulation in the reservoir.

3.2 Piston compressor and compressed air reservoir

Air compression and storage is an unsteady process due to the air pressure in the compressed air reservoir varying over time. Hence, the power consumption of the piston compressor has to be taken into account. The piston compressor is mechanically connected to the diesel engine and as a result operates at a constant rotational speed and mass flow rate when clutch 1 is on. As a general observation in multistage piston compressor, pressure ratio of initial stage (low pressure stages) is higher compare to final stage (high pressure stage). However in a preliminary study an equal compression ratio model is adequately accurate [25]. Therefore for a N -stage piston, assuming the compression ratio is the same for each stage and the compression is polytropic, the power consumption W_{PC} is then calculated by:

$$W_{PC} = \dot{m}_{PC} \cdot \frac{N \cdot n}{n-1} R T_{am} \left[\left(\frac{P_S}{P_{am}} \right)^{\frac{n-1}{n \cdot N}} - 1 \right] \quad \text{(Eq. 9)}$$

In equation 9 $\dot{m}_{PC}$ and N represent the mass flow rate of air and the stage number of compression with inter-cooling, R is the universal gas constant, n is the polytropic factor which has a value ranging between 1.0 and γ (the adiabatic index) with $n = 1.0$ being the isothermal process and $n = \gamma$ the adiabatic process, P_{am} and T_{am} stand respectively for the ambient pressure and temperature, and P_S is the pressure in the compressed air reservoir. This study uses data from available commercial compressors with a polytrophic factor of $n = 1.25$ and a stage number of $N = 4$. It is worth noting that from modeling point of view the compressor operates with time variable compression ratio since pressure in the compressed air reservoir changes over time.

It is also assumed that the temperature of the compressed air in the reservoir is constant which equals to the ambient temperature. Such an assumption is reasonable due to inter-cooling in the multistage compression process. And furthermore compression heat will also release to surroundings in the storage period. Based on this assumption the process of compressing the air into the reservoir, or releasing compressed air from the reservoir, affects only the storage pressure P_S. This can be calculated from the total mass of compressed air and the volume of the reservoir. It's also worth mentioning that in this study we only consider isochoric storage instead of isobaric storage which ideally is more efficient but less developed [10].

3.3 Air turbine and heat exchangers

Similar to the inlet compressor, the air turbine has to operate under off-design conditions in order to respect the power requirement of the inlet compressor (operational modes 1 and 3) or end-users (operational mode 5). Similarly it is assumed that the expansion processes in high-pressure stage and low-pressure stage of the air turbine have the same pressure ratio. In a rotary turbine the mass flow rate depends on the inlet conditions and the corrected mass flow rate of the air turbine under off-design conditions is expressed as:

$$\frac{\dot{m}_{AT}}{\dot{m}_{AT}^{rated}} = \frac{P_{AT}}{P_{AT}^{ref}} \sqrt{\frac{T_{AT}^{ref}}{T_{AT}}} \qquad \text{(Eq. 10)}$$

In equation 10 P_{AT}^{ref} and T_{AT}^{ref} denote respectively, the reference inlet pressure and temperature (values at the design point) while P_{AT} and T_{AT} are the real-time inlet pressure and temperature. The isentropic efficiency of the air turbine and the generated power can then be calculated similarly according to the inlet compressor. Eq. (10) is applied separately to both high pressure turbine and low pressure turbine. Table 3 lists the design conditions for the two turbines;

nominal inlet temperature is 200°C while nominal expansion ratio is 10 for both turbines. Inlet pressure for the high pressure turbine is the same as CAES pressure at each given time.

It is worth mentioning that the heat exchangers are used for both heat transfer and thermal energy storage. In cases of operational modes 1 and 3, the flue gas and the compressed air exchange heat directly in conventional ways. In cases of operational modes 2 and 4 the thermal energy of the flue gas is stored within the PCMs of the heat exchangers and, the stored thermal energy is recovered in Mode 5,when the diesel engine is turned off. It is also worth mentioning that under design conditions the air turbine is coupled with the inlet compressor to drive the air pre-compression process. As mentioned, due to the much higher expansion ratio in air turbine and much lower compression ratio in inlet compressor, the rated mass flow rate in air turbine is generally a forth to a fifth of that in inlet compressor. In another word the waste heat generated in diesel engine is much more than the required heat in air turbine in modes 1 and 3. The excessive heat, as well as the waste heat generated in modes 2 and 4, are stored in the PCMs which is quantitatively adequate to supply heat to air turbine in mode 5. From our modelling study the longest contiuous period of mode 5 operation is 10 hours. Thus the size of PCM storage should guarantee a continuous heat supply for such period of time. it results that using a phase change material with energy storage density of 200 kJ/kg (molten salt) the storage system has to accommodate about 0.5 m^3 of PCM.

3.4 Balance of the system

The problem to be resolved in this study consists of finding the correct operational mode and the operational state for a given load and real-time states (pressures) of compressed air. Thus the following equations are verified:

1) Balance equation of the diesel engine’s crankshaft

The power supplied by the diesel engine (either supercharged or non-charged) must be equal to the total load requirement, including the power consumption of the piston compressor:

$W_{SD} = W_{EU}$ for Mode 1;

$W_{DE} = W_{EU}$ for Mode 2;

$W_{SD} = W_{EU} + W_{PC}$ for Mode 3;

$W_{DE} = W_{EU} + W_{PC}$ for Mode 4.

2) Balance equation of the air turbine

The power generated by the air turbine must be equal to the power consumption of the inlet compressor or the power requirement of the external end-users:

$W_{AT} = W_{IC}$ for Modes 1 and 3;

$W_{AT} = W_{EU}$ for Mode 5.

As described above the power output, together with the mass flow rate of the air turbine, relates to the pressure ratio. The inlet pressure of the high-pressure air turbine in the system could be adjusted using the valve located at the outlet of the compressed air reservoir. This should be lower than either the reference value or the pressure in the reservoir.

$$P_{AT,H} < \min\left(P_{AT,H}^{ref}, P_S\right)$$

The inlet temperature of the air turbine should also lower than both the design value and the temperatures of the flue gas or the PCMs in the heat exchangers. This is physically achievable by the design of a multi-channel heat exchanger: in modes 1 and 3 the high pressure air exchanges heat with flue gas and in mode 5 the high pressure air exchanges heat with PCMs.

$T_{AT} = T_{ex} - \Delta T_{loss}$ for Modes 1 and 3;

$T_{AT} = T_{PCMs} - \Delta T_{loss}$ for Mode 5.

In the above equation T_{ex} and T_{PCMs} are the temperatures of flue gas and the PCMs medium. ΔT_{loss} is the temperature loss in the heat transfer process.

3) Balance equation of the compressed air reservoir

In anisometric storage system the balance of compressed air in the reservoir is governed by the following equation:

$$\left(\dot{m}_{PC} - \dot{m}_{AT}\right) \cdot \Delta t = \left(\rho_S^{t+\Delta t} - \rho_S^{t}\right) \cdot V_S \qquad \text{(Eq. 11)}$$

In equation 11 t and Δt are the operational time and time interval respectively, V_S is the volume of the compressed air reservoir, $\dot{m}_{PC}$ and $\dot{m}_{PC}$ are the piston compressor mass flow rate and the air turbine mass flow rate, ρ_S^{t} and $\rho_S^{t+\Delta t}$ respectively are the density of compressed air before and after the time-step operation. This can be calculated from the corresponding storage pressures based on the isothermal assumptions.

The equations presented in Sect. 3 were implemented in Matlab 2014®. The equations constitute a close set of equation that was solved at each instant of time to evaluate all the variable of interests including mass flow rates, pressures, temperatures and electric power.

4. System performance evaluation and discussion

The diesel-CAES integrated system is proposed to generate electricity for domestic users in remote areas. A small isolated village of 100 households is used as an example for the external end-users. The individual household electricity consumption recorded with 5-minutes intervals in Newcastle (England) and Llanelli (Wales) is selected as the data resources of elementary electricity consumption profiles. However in this study we reset the time interval to be 2 hours by averaging the numbers within the period. This is because in real applications once the diesel engine is started, it should remain in service for a minimum amount of time of at least one hour [26, 27] and short-term intermittence should be met by power quality improvement technologies such as rechargeable batteries.

Figure 3 shows a one-year electricity consumption profile of the isolated village on a day-to-day basis starting from the first of January. This reveals that the power consumption in winter is higher than in summer due to space heating. Figure 4 indicates accordingly, the electricity requirement distribution based on a 2-hour interval. It is found that the power requirement was mainly in the region of 20kW to 80kW. However, the maximum electricity requirement is as high as 180kW. As a result the high capacity generation has to be installed to supply the peak demand, but will ultimately be idle most of the time. It should be noted in Figure 3, the average daily electricity requirement shows the peak value is lower than those based on a 2-hour interval in Figure 4.

The conventional generation capacities of two different schemes with diesel-only engines are used as benchmarks to evaluate and compare the performance of the diesel-CAES integrated system. Three systems considered in this study are:

- System 1: This system is a non-charged diesel engine. In order to produce electricity for the end-users independently, the rated capacity of the engine is set to equal the maximum load requirement for a full year $W_{DE}^{\max} = W_{EU}^{\max}$.
- System 2: This system is made up of two non-charged diesel engines. One is used as the base capacity while the other is used as the peak load. The rated capacity of the base load engine is equal to the average load of the end-users $W_{DE1}^{\max} = \overline{W}_{EU}$ where $\overline{W}_{EU}$ is the annual average load of the end-users. And the peak load engine helps cover the maximum load requirement with the rated capacity of $W_{DE2}^{\max} = W_{EU}^{\max} - \overline{W}_{EU}$ (generally $W_{DE2}^{\max} > W_{DE1}^{\max}$). This system operates based on the following rules: in the case that the

end-users' load is lower than the average load, only the base load engine works. Else when the end-users' load is lower than the peak engine's rated capacity, only the peak engine works. Otherwise while the end-users' load is higher than the peak engine's rated capacity, both the engines are turned on to accommodate the end-users' demand. It should be noted that although theoretically more diesel engines with different capacity can be adopted, it causes much more frequent starting and stopping which has negative effects on the efficiency and lifetime of the engine [3].

- System 3: This is the diesel-CAES integrated system. The maximum electricity output of this system with supercharging is set as $W_{SD}^{\max} = \phi \cdot W_{EU}^{\max}$ where $\phi = 1.05$ is the safety coefficient. This is because the maximum power output of the integrated system depends on the storage pressure of the compressed air. If the storage pressure P_S is lower than the reference pressure of air turbine, the maximum power output of the integrated system is then lower than the rated value. Apart from the scale of the engine and the settings described in Section 3, the other key parameters for the integrated system are listed in Table 3.

The fuel consumption rate and efficiency (diesel engine efficiency for system 3) of the three systems are shown respectively in Figures 5 and 6 (in order to make the images clearer, only one point is plotted per day). It demonstrates that the idling fuel consumption plays an important role in the performance of the diesel engine. System 1 has the biggest scale diesel engine and, as a result, the largest idling fuel consumption. This leads to a fuel consumption rate of 22 L/hour or higher even at idling or low load factor operation, resulting in the engine operating at an efficiency that is lower than 20% for the majority of the time. As shown in the figures, Systems 2 using two engines as an alternative is an efficient way to save fuel. As shown in Figure 5 the two diesel engines in System 2 operate simultaneously while the electricity demand is higher than the single engine's rated power. During these times, fuel consumption and efficiency are almost the same as System 1. However, when the end-users' demand is lower than the single engine's rated power, only one engine is used. Consequently the fuel rate decreases lower than 15 L/hour, enabling efficiency to be maintained in excess of 20%. This advantage avoids the diesel engine operating at very low load factors and, as a result, significantly improves the overall performance of the system.

Compared with System 2, System 3 enhances performance by removing the idling fuel consumption at lower demands. As seen in Figure 5, the diesel engine is turned off (operational mode 5) for a considerable part of the operating times when the end-users' demand is powered solely by the air turbine. This keeps the diesel engine operating at only high load factors and, as a result, the efficiency is always more than 25%. Figure 6 illustrates the high efficiency of the engine in System 3, which is often as high as 53% when the diesel engine is supercharged. It should be noted that this efficiency, as defined in Section 3, is higher than the traditional thermal efficiency. This is becasue the diesel engine is externally powered by the air turbine. The overall fuel consumption of the diesel-CAES integrated system is far less than Systems 1 and 2, as shown in Figure 7. This results in the annual diesel fuel consumption of the three systems being 242m^3, 158m^3 and 121 m^3, respectively. Thus, using System 2 to replace System 1 leads to a reduction in fuel of 34.7%. Moreover, using system 3 to replace system 2 brings a further 23.4% reduction, thanks to the integration of the compressed air energy storage unit.

Overall, one can see that system 1 has only one operational mode, while system 2 and system 3 have 3 and 5 operational modes respectively. The more operational modes the system has, the higher the efficiency it works at to keep variable load profiles consistent. Figure 8 plots the operational modes of system 3 at varying times. Apart from the most frequent operational modes 5 and 2, operational modes 3 and 4 account for a large part of the overall operation. The participation of operational modes 3 and 4 changes the distribution pattern of the electricity generated by the diesel engine, as illustrated in Figure 9. When comparing with Figure 4, the load of the diesel engine below 50 kW is removed, partially by operational modes 3 and 4 to a region higher than 100 kW. This indicates that the piston compressor is an additional help to the diesel engine when operating at very high load factors and, as a result, enhances the overall performance of the system.

The overall performance of the CAES unit is important to the system. Because of the energy losses in the compression and expansion processes, the isolated CAES (without combustion) is restricted due to its low efficiency. However, in the diesel-CAES integrated system this disadvantage is overcomed by the recovery of the waste heat in the flue gas. Figure 10 shows the overall energy generation of the diesel engine in three systems. It is found the net power generation of the diesel engine in system 3 is 408 MWh, which is lower than those of system 1 and 2 (419 MWh). Therefore, the overall power generation of the air turbine is greater than the overall power consumption of the piston compressor. This results in an effective energy storage efficiency (the ratio of energy generated by air turbine and energy consumed by piston compressor) of more than 100% due to the use of heat recovery. This is easily understandable due to the high temperature of the flue gas being 200°C to 500°C or even higher. From the thermodynamic view, heating the compressed gas from ambient to 300 °C prior to expansion roughly doubles the net power output. In particular, as the air expansion is an open cycle the compressed air can be heated up as high as possible even with variable flue gas, making the heat recovery process quite efficient.

The advanced pressure storage vessel is always an issue for small scale CAES as it is costly to develop and to safety-test. Therefore a smaller volume of the advanced pressure vessel makes the diesel-CAES integrated system more competitive from an economic aspect. In the above example, the volume of the vessel is set at $10m^3$ with the real-time storage pressure illustrated in Figure 11. Increasing the vessel volume will decrease the pressure fluctuation as shown in Figure 12, while based on the simulation changes of the fuel consumption are negligible (the annual average efficiency increases from 34.7% to 34.9% when changing the storage volume from $10m^3$ to $20m^3$). However by decreasing the storage volume to a lower than critical value this may cause failure of the electricity supply as the peak load generation (operational mode 1) requires continuous compressed air supply. The time interval is the key parameter affecting the critical storage volume. Generally the decrease in the time interval results in a proportional decrease of the critical storage volume. For example the critical storage volumes for system 3 above are respectively $10m^3$, $4.75m^3$ and $2.8m^3$ for the time intervals of 2 hours, 1 hour and 0.5 hour. However, it should be noted that in the above simulation the switching time of different operational modes are ignored, while in practice the heating up of compressed air may take several minutes or even longer. Furthermore, the frequent switch of the operational mode also shortens its lifetime as mentioned. As a result, selection of the storage volume is a balance of contributing factors mentioned above, together with the costs of power quality improvement and load following facilities.

As discussed above, the diesel-CAES integrated system in this study is proposed to efficiently generate electricity for isolated end-users with highly variable load patterns. Therefore the characteristics of the load pattern play a pivotal role in assessing the performance of the system. The load patterns of the end-users vary significantly with the location, the users' composition, economics and many other factors. However in this first step of a feasibility study a statistical parameter named 'Normalized standard deviation' is used to roughly evaluate the characteristics of the load profile defining as:

$$\sigma = \sqrt{\frac{1}{N}\sum_{t=1}^{N}\left(\frac{W_{EU}^{t}}{W_{EU}^{\max}} - \frac{\overline{W}_{EU}}{W_{EU}^{\max}}\right)^{2}} \qquad \text{(Eq. 12)}$$

In equation 12 W_{EU}^{t} is the end-users' load in a specific time (in this study the average load is within 2 hours), $N = 4380$ is the number of timed intervals in a full year. From this definition one can see the specific load is normalized first by dividing the max load in the year $W_{EU}^{\max}$ and then the standard deviation is calculated in a normal way. As a result the normalized standard deviation measures the normalized spread of the specific load when calculating the yearly mean load.

Using the example load profile in Figure 3 as the baseline, different load profiles are developed numerically by changing the altitude and the base-load. If these developed loads are supplied independently by the three systems discussed above, the annual average efficiencies are plotted in Figure 13. It is found the efficiency of system 1 decreases linearly with the increase of the normalized standard deviation, indicating the single engine is not suitable for a highly variable load. In comparison, system 2 can efficiently cover slightly changing load profiles with the normalized standard deviation lower than 0.14, otherwise the annual average efficiency decreases linearly as system 1. For system 3 the result is adverse. When the load is very stable with the normalized standard deviation lower than 0.14, the annual average efficiency decreases while the normalized standard deviation goes down. Alternatively, the annual average efficiency keeps a high level if the normalized standard deviation is higher than 0.14. This again suggests that system 3 has a greater advantage, particularly for highly variable end-users, in particular for the cases with high share of wind power generation. Of course it should be noted that the efficiency of system 3 is always much higher than system 1 and system 2, in part thanks to the efficient recovery of the waste heat in the flue gas.

5. Conclusions and future work

This paper proposes a diesel-CAES integrated system to supply electricity for isolated end-users such as a remote village. The integrated system has five operational modes with the operational principles developed accordingly. Using a single diesel system and a dual-diesel system as baselines, the system performance is numerically studied to power a UK small-scale village solely. The results show the fuel consumption of the integrated system is only 50% of the single diesel system and 77% of the dual-diesel system. Meanwhile, the volume of the high-pressure vessel for such an integrated system is found to be feasible approximately 5m^3 for use in a small village with an interval time of 1 hour. The characteristics of the end-users' load pattern is also studied using a statistical parameter named 'Normalized standard deviation' and shows the integrated system performs very well, particularly for highly variable load patterns or for high share wind power generation. The authors are currently working on the construction of a lab-scale pilot system and the results will be reported in the near future.

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