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Integrating compressed air energy storage with a diesel engine for electricity generation in isolated areas

Li, Yongliang; Sciacovelli, Adriano; Peng, Xiaodong; Radcliffe, Jonathan; Ding, Yulong

Link to publication on Research at Birmingham portal

1 **INTEGRATING COMPRESSED AIR ENERGY STORAGE WITH A DIESEL**

2 **ENGINE FOR ELECTRICITYGENERATION IN ISOLATED AREAS**

3 Yongliang Li[*], Adriano Sciacovelli, Xiaodong Peng, Jonathan Radcliffe and Yulong Ding

4 School of Chemical Engineering, University of Birmingham, Birmingham B15 2TT, UK

5 **Abstract**

6 This paper reports an integrated system consisting of a diesel genset and a Compressed Air

7 Energy Storage (CAES) unit for power supply to isolated end-users in remote areas. The

8 integration is through three parts: direct-driven piston-compression, external air turbine-driven

9 supercharging, and flue gas waste recovery for super-heating. The performance of the

10 integrated system is compared for a single diesel unit and a dual-diesel unit with a capacity of

11 electricity supply to a village of 100 households in the UK setting. It is found the fuel

12 consumption of the integrated system is only 50% of the single-diesel unit and 77% of the

13 dual-diesel unit. The addition of the CAES unit not only provides a shift to electrical energy

14 demand, but also produces more electricity due to the recovery of waste heat.

15 **Key words:** district energy supply; compressed air energy storage; thermal energy storage;

16 supply side management; system integration.

17

18 [*]Corresponding author: tel.+44 (0)121 414 5135; e-mail y.li.1@bham.ac.uk

19

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 (-)

20

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

46 to useful work, energy transferred to coolant and energy lost through exhaust [13, 14].

47 Thermal energy loss from the exhaust can be regarded as a high grade, which has a

48 temperature ranging approximately from 400 to 600°C [15]. Recent work has shown a

49 potential increase in the overall efficiency by up to 30% through efficient recovery of waste

50 heat [16]. Technologies proposed for the recovery of waste heat include Organic Rankine

51 Cycle (ORC) [17], thermoelectric generation [18] and the use of heat pumps [19]. However,

52 when a diesel engine is used for remote electricity generation, the temperature of the exhaust

53 gas changes frequently as does the load factor [20]. Thermoelectric power generation is

54 expensive and has a low efficiency. The unsteady exhaust gas temperature is disadvantageous

55 for the operation of an ORC engine or a heat pump. Compressed Air Energy Storage (CAES)

56 presents an alternative solution to the issue, which can store excessive shaft power, and

57 recover the waste heat of the diesel engine in the energy extraction process. Using CAES to

58 deal with the stochastic fluctuations of wind power in wind-diesel hybrid systems has been

59 examined numerically, and the results are promising in enhancing the wind energy penetration

60 [2, 21]. In this paper, an integrated diesel-CAES power system is proposed and investigated.

61 The aim is to reduce fuel consumption and production costs for electricity generation in rural

62 areas. Specific attention is paid to the operating principle and the influence of demand patterns

63 of end-users. This may lead to a real system to be developed accordingly to demonstrate the

64 advantages.

65

66 **2. System configuration and operating principle**

67 Most modern diesel engines are turbocharged or even supercharged. A turbocharger or a

68 supercharger is made up of a coupled compressor-turbine unit aiming to increase the density

69 of the engine air intake. This results in the engine producing significantly more power than a

70 naturally aspirated engine with the same combustion-chamber volume. The difference

71   between the turbocharger and supercharger is that the supercharger has a compressor driven

72   mechanically by external power such as the engine's crankshaft, while a turbocharger is

73   powered by the engine exhaust, therefore does not require any mechanical power.

74

75   This paper focuses on an integrated diesel-CAES system in which the diesel engine could be

76   supercharged by a CAES unit, as illustrated in Figure 2. The diesel engine used in such an

77   integrated system differs from the traditional engine in the air intake method: the atmospheric

78   air can either be naturally aspirated or forcibly compressed into the combustion-chamber

79   depending on the switch conditions of a 3-way valve located at the engine inlet. The CAES

80   unit is made up of a piston compressor, a compressed air reservoir, two heat exchangers and a

81   two-stage air turbine. The diesel engine and the CAES unit are integrated by three parts: first

82   the diesel engine shaft and the piston compressor shaft could be mechanically connected or

83   disconnected through using Clutch 1. Second, the air turbine shaft in the CAES unit and the

84   compressor shaft in the diesel engine can be mechanically connected or disconnected by the

85   use of Clutch 3. Third, the flue gas from the diesel engine and the compressed air from the

86   reservoir are both fed into Heat exchangers 1 and 2 for waste heat recovery. It is worth

87   mentioning that two heat exchangers are used not only for the heat transfer between the flue

88   gas and compressed air, but also for the storage of thermal energy in cases where the engine

89   and air turbine operate at different times. Phase Change Materials (PCMs) can be packed into

90   the heat exchangers for high-density thermal energy storage. Examples of PCMs for such an

91   application are composite materials consisting of an inorganic salt (PCM) and a ceramic

92   matrix due to their favorable costs, good energy density and a wide range of melting

93   temperatures [22].

94

95   From the above one can see there are three power-related components in the integrated system:

96   diesel engine, piston compressor and air turbine (includes the coupled compressor for

97   supercharging). Theoretically, based on the operating status of the three components, the

98   system has 8 operating modes, as listed in Table 1.

99

100   Of the modes shown in Table 1, Mode 6 is the off state of the integrated system whereas Mode

101   7 is almost inapplicable as the air turbine is not connected to the piston compressor in the

102   system. In addition, from an energy utilization point of view, Mode 7 is not practical as it

103   produces nothing, but consumes compressed air, due to process irreversibility. Similarly,

104   Mode 8 is virtually impossible as no power is available to drive the piston compressor. As a

105   result, the integrated diesel-CAES system has the following five potentially useful operating

106   modes:

107   • Mode 1 focuses on a supercharged-diesel process to respect the high demand of end-

108   users. In this process Clutch 3 is connected while Clutch 1 and Clutch 2 are

109   disconnected. The 3-way valve is turned towards the compressor side so that the

110   coupled air turbine-compressor is able to intake more air to be compressed into the

111   diesel chamber. The reason for using a coupled air turbine-compressor instead of

112   charging the diesel chamber directly with compressed air from the reservoir, is that the

113   air pressure in the reservoir is much higher than the required pressure of diesel

114   chamber. With waste heat recovery from flue gas the high-pressure compressed air in

115   the reservoir can drive the coupled air turbine-compressor to produce about 5 times of

116   low-pressure compressed air that is required by the diesel chamber.

117   • Mode 2 is a mode with all the clutches disconnected and the 3-way valve turned

118   towards the atmospheric side. As a result the diesel engine runs in a traditional manner:

119   the atmospheric air is naturally aspirated into the cylinder for combusting the diesel

120   fuel. The shaft power produced by the diesel engine is used to generate electricity for

121   end-users. The exhaust gas of the diesel engine is used to heat the PCMs in Heat

122   Exchangers 1 and 2 in order to recover the high grade heat for later uses in the

123   compressed air expansion process.

124   •   Mode 3 is similar to Mode 1 but with Clutch 1 connected. The power generated by the

125   supercharged diesel engine is used to respect the end-users' demand, as well as drive

126   the piston compressor to produce the compressed air.

127   •   Mode 4 is similar to Mode 2 but with Clutch 1 connected. The power generated by the

128   diesel engine is used to respect the end-users' demand as well as drive the piston

129   compressor to produce the compressed air.

130   •   Mode 5 is a case with the diesel engine turned off, Clutch 1 and Clutch 3 disconnected

131   and Clutch 2 connected. The compressed air is heated up first by the thermal energy

132   stored in Heat exchanger 1 and Heat exchanger 2 prior to the expansion in the air

133   turbines to drive Generator 2 to produce electricity for end-users. Such a process

134   avoids the diesel engine to operate at a very low load factor and as a result saves on

135   fuel consumption.

136

137 The proposed integrated diesel-CAES system is designed to match the load of typical end-

138 users in remote areas without access to electric grids. Therefore, the operation mode of the

139 system has to be updated regularly after each operating step. In the operational mode selection

140 process, not only the end-user's demand, but also the pressure of the air reservoir and the

141 status of the heat stored in the heat exchangers play decisive roles. In this study a control

142 algorithm is developed and programmed in MATLAB. In this program, the power

143 consumption of the piston engine, the maximum power outputs of the air turbine and the

144 supercharged diesel engine are calculated based on the updated inputs. Table 2 presents the

145 logic to select the operation mode at each instant of time. First pressure P_S in the CAES

146 reservoir and end user demand W_{EU} are assessed. If P_S is higher than the maximum allowed

147 value CAES charging is not possible; otherwise CAES charging can potentially take place in

148 the case power output of the asset is large enough to satisfy both the end user demand and

149 provide power to the piston compressor to charge the CAES. If this is not the case meet the

150 satisfying the end user demand has the priority over CAES charging. When storage pressure

151 is within the allowed range ($P_S^{min} < P_S < P_S^{max}$) CAES discharge can occur and if the end user

152 demand is smaller than the power delivered by the air turbines. It should be noted that such a

153 selection principle is based on the assumption that the rated mass flowrate of the piston engine

154 is higher than that of the air turbine. This ensures the possibility to charge the CAES reservoir.

155 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

156 withdrawn from the CAES would be higher that the injected one without the necessary

157 conditions for air accumulation in the reservoir.

158

159 Compared to non-charged diesel engines, the integrated diesel-CAES system can downsize the

160 scale of the facility due to the application of supercharging, thus enabling the system to be

161 operated at a high load factor. In addition, the integrated system could supply electric power

162 solely by air turbine when the end-users' demand is low (Mode 5), thus avoiding the diesel

163 engine to be operated at a very low load factor. In the following, attention is given to fuel

164 consumption of the integrated diesel-CAES system using the results of traditional diesel sets

165 as the baseline.

166

167 **3. Thermodynamic modelling of the key processes**

168   Numerical modeling is employed in this section to examine the effect of the use of CAES on

169   the efficiency and fuel consumption of the diesel engine. As described in Section 2, the

170   integration is through 3 main parts in the diesel-CAES integrated system consisting of the

171   diesel engine, the piston-compressor and compressed air reservoir, the air turbines (including

172   the heat exchangers) and a coupled compressor. In the following, each of the components will

173   be numerically modeled to evaluate the overall performance of the integrated system. It should

174   be noted that many factors influence the performance of the components including the

175   manufacturer, the operational conditions, and sizes etc. However, as this work represents a

176   first step towards developing such an integrated system, generic models are adopted.

177

178   3.1 Diesel engine (including the inlet compressor)

179   This study focuses on two main performance indicator for the diesel engine: the efficiency and

180   the fuel consumption. In a diesel engine the fuel consumption rate is governed mechanically or

181   electronically by a fuel injection system to meet the required load factor. In our study the

182   diesel engine works in two different modes depending on the end-users' demand. When the

183   demand is lower than the rated power (the maximum power output without supercharging), the

184   supercharging unit is switched off (Mode 2 and 4). Therefore the fuel consumption and

185   thermal efficiency of the engine could be estimated as[6, 23]:

186
$$\frac{\dot{m}_f}{\dot{m}_f^{rated}} = \xi + (1 - \xi) \cdot \frac{W_{DE}}{W_{DE}^{max}}$$
(Eq. 1)

187
$$\eta_{DE} = \frac{W_{DE}}{\dot{m}_f \cdot Q_{LHV}}$$
(Eq. 2)

188   In the above equations, $\dot{m}_f$ and $\dot{m}_f^{rated}$ are respectively the real-time and rated flow rate of the

189   diesel fuel, W_{DE} and W_{DE}^{max} are respectively the real-time and rated power output, ξ is a

190   constant related to the consumption curve of the generator and is equal to 0.34 for a non-

191   charging diesel engine, Q_{LHV} is the lower heating value of diesel fuel equal to 43.4 MJ/kg.

192   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

193   study. As the diesel engine operates at a constant rotational speed for electricity generation,

194   the mass flow rate is considered to also be constant in this mode with the rated air/fuel ratio

195      $\lambda^{rated} = \dfrac{\dot{m}_a^{rated}}{\dot{m}_f^{rated}}$ equal to 14.7 based on stoichiometric balance.

196

197   When the end-users' demand is higher than the rated power, the supercharging unit is

198   switched on so more air can be blown into the combustion-chamber by the inlet compressor

199   (operational mode 1/3). For a constant speed diesel engine with pre-cooling, the mass of air

200   entering the engine is proportional to the inlet pressure as:

$$\frac{\dot{m}_a}{\dot{m}_a^{rated}} = \frac{P_{IC}}{P_{am}}$$

201   (Eq. 3)

202   In the above equation P_{IC} is the outlet pressure of the inlet compressor, P_{am} is the ambient

203   pressure, and $\dot{m}_a$ is the mass flow rate of air entering the engine The power consumption of the

204   compressor is featured with isentropic efficiency. This is the comparison between the actual

205   performance and the performance that would be achieved under idealized circumstances

206   (isentropic processes) for the same inlet state and the same exit pressure. In order to meet the

207   users' demand at all times, the compressor has to operate under off-design conditions using

208   different outlet pressure and flow rate. The isentropic efficiency η_{IC} of the inlet compressor

209   varies with the real-time outlet pressure and can be evaluated as [24]:

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

210   (Eq. 4)

211   In the above equation Δh_s indicates the enthalpy change in an isentropic process, while the

212   superscript ref denotes the reference state at the design point or process. The necessary work

213   for compression then can be expressed as:

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

215   In the above equation W_s is the specific power consumption in an isentropic process and W_{IC}

216   is the actual power required for the compression process

217   The thermal efficiency η_{SD} of the diesel engine relates to the real-time air/fuel ratio $\lambda = \dfrac{\dot{m}_a}{\dot{m}_f}$

218   following a quadratic model [2]:

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

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

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

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

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

224   In the above equation T_{am} is the ambient temperature. A and B are constants which equal to

225   1000K and 0.0667 based on the experimental tests.

226   It should be noted that such a selection principle is based on the assumption that the rated

227   mass flowrate of the piston engine is higher than that of the air turbine. This ensures the

228   possibility to charge the CAES reservoir. In fact, if $\dot{m}_{PC} \neq 0$; $\dot{m}_{AT} \neq 0 \neq 0$ and $\dot{m}_{PC} < \dot{m}_{AT}$

229   under design conditions the mass flow rate withdrawn from the CAES would be higher that

230   the injected one without the necessary conditions for air accumulation in the reservoir.

231

232

233   3.2 Piston compressor and compressed air reservoir

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

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

244   In equation 9 $\dot{m}_{PC}$ and N represent the mass flow rate of air and the stage number of

245   compression with inter-cooling, R is the universal gas constant, n is the polytropic factor

246   which has a value ranging between 1.0 and γ (the adiabatic index) with $n = 1.0$ being the

247   isothermal process and $n = \gamma$ the adiabatic process, P_{am} and T_{am} stand respectively for the

248   ambient pressure and temperature, and P_S is the pressure in the compressed air reservoir. This

249   study uses data from available commercial compressors with a polytrophic factor of $n = 1.25$

250   and a stage number of $N = 4$. It is worth noting that from modeling point of view the

251   compressor operates with time variable compression ratio since pressure in the compressed air

252   reservoir changes over time.

253

254   It is also assumed that the temperature of the compressed air in the reservoir is constant which

255   equals to the ambient temperature. Such an assumption is reasonable due to inter-cooling in

256   the multistage compression process. And furthermore compression heat will also release to

257   surroundings in the storage period. Based on this assumption the process of compressing the

258   air into the reservoir, or releasing compressed air from the reservoir, affects only the storage

259   pressure P_S. This can be calculated from the total mass of compressed air and the volume of

260   the reservoir. It's also worth mentioning that in this study we only consider isochoric storage

261   instead of isobaric storage which ideally is more efficient but less developed [10].

262

263   3.3 Air turbine and heat exchangers

264   Similar to the inlet compressor, the air turbine has to operate under off-design conditions in

265   order to respect the power requirement of the inlet compressor (operational modes 1 and 3) or

266   end-users (operational mode 5). Similarly it is assumed that the expansion processes in high-

267   pressure stage and low-pressure stage of the air turbine have the same pressure ratio. In a

268   rotary turbine the mass flow rate depends on the inlet conditions and the corrected mass flow

269   rate of the air turbine under off-design conditions is expressed as:

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

271   In equation 10 P_{AT}^{ref} and T_{AT}^{ref} denote respectively, the reference inlet pressure and temperature

272   (values at the design point) while P_{AT} and T_{AT} are the real-time inlet pressure and temperature.

273   The isentropic efficiency of the air turbine and the generated power can then be calculated

274   similarly according to the inlet compressor. Eq. (10) is applied separately to both high pressure

275   turbine and low pressure turbine. Table 3 lists the design conditions for the two turbines;

276 nominal inlet temperature is 200°C while nominal expansion ratio is 10 for both turbines. Inlet

277 pressure for the high pressure turbine is the same as CAES pressure at each given time.

278

279 It is worth mentioning that the heat exchangers are used for both heat transfer and thermal

280 energy storage. In cases of operational modes 1 and 3, the flue gas and the compressed air

281 exchange heat directly in conventional ways. In cases of operational modes 2 and 4 the

282 thermal energy of the flue gas is stored within the PCMs of the heat exchangers and, the stored

283 thermal energy is recovered in Mode 5,when the diesel engine is turned off. It is also worth

284 mentioning that under design conditions the air turbine is coupled with the inlet compressor to

285 drive the air pre-compression process. As mentioned, due to the much higher expansion ratio

286 in air turbine and much lower compression ratio in inlet compressor, the rated mass flow rate

287 in air turbine is generally a forth to a fifth of that in inlet compressor. In another word the

288 waste heat generated in diesel engine is much more than the required heat in air turbine in

289 modes 1 and 3. The excessive heat, as well as the waste heat generated in modes 2 and 4, are

290 stored in the PCMs which is quantitatively adequate to supply heat to air turbine in mode 5.

291 From our modelling study the longest contiuous period of mode 5 operation is 10 hours. Thus

292 the size of PCM storage should guarantee a continuous heat supply for such period of time. it

293 results that using a phase change material with energy storage density of 200 kJ/kg (molten

294 salt) the storage system has to accommodate about 0.5 m^3 of PCM.

295

296 3.4 Balance of the system

297 The problem to be resolved in this study consists of finding the correct operational mode and

298 the operational state for a given load and real-time states (pressures) of compressed air. Thus

299 the following equations are verified:

300   1) Balance equation of the diesel engine's crankshaft

301   The power supplied by the diesel engine (either supercharged or non-charged) must be equal

302   to the total load requirement, including the power consumption of the piston compressor:

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

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

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

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

307   2) Balance equation of the air turbine

308   The power generated by the air turbine must be equal to the power consumption of the inlet

309   compressor or the power requirement of the external end-users:

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

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

312   As described above the power output, together with the mass flow rate of the air turbine,

313   relates to the pressure ratio. The inlet pressure of the high-pressure air turbine in the system

314   could be adjusted using the valve located at the outlet of the compressed air reservoir. This

315   should be lower than either the reference value or the pressure in the reservoir.

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

317   The inlet temperature of the air turbine should also lower than both the design value and the

318   temperatures of the flue gas or the PCMs in the heat exchangers. This is physically achievable

319   by the design of a multi-channel heat exchanger: in modes 1 and 3 the high pressure air

320   exchanges heat with flue gas and in mode 5 the high pressure air exchanges heat with PCMs.

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

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

323 In the above equation T_{ex} and T_{PCMs} are the temperatures of flue gas and the PCMs medium.

324 ΔT_{loss} is the temperature loss in the heat transfer process.

325   3) Balance equation of the compressed air reservoir

326 In anisometric storage system the balance of compressed air in the reservoir is governed by the

327 following equation:

328 $$\left(\dot{m}_{PC} - \dot{m}_{AT} \right) \cdot \Delta t = \left(\rho_S^{t+\Delta t} - \rho_S^t \right) \cdot V_S \qquad \text{(Eq. 11)}$$

329 In equation 11 t and Δt are the operational time and time interval respectively, V_S is the

330 volume of the compressed air reservoir, $\dot{m}_{PC}$ and $\dot{m}_{PC}$ are the piston compressor mass flow

331 rate and the air turbine mass flow rate, ρ_S^t and $\rho_S^{t+\Delta t}$ respectively are the density of

332 compressed air before and after the time-step operation.   This can be calculated from the

333 corresponding storage pressures based on the isothermal assumptions.

334 The equations presented in Sect. 3 were implemented in Matlab 2014®. The equations

335 constitute a close set of equation that was solved at each instant of time to evaluate all the

336 variable of interests including mass flow rates, pressures, temperatures and electric power.

337

338 **4. System performance evaluation and discussion**

339 The diesel-CAES integrated system is proposed to generate electricity for domestic users in

340 remote areas. A small isolated village of 100 households is used as an example for the external

341 end-users. The individual household electricity consumption recorded with 5-minutes intervals

342 in Newcastle (England) and Llanelli (Wales) is selected as the data resources of elementary

343 electricity consumption profiles. However in this study we reset the time interval to be 2 hours

344 by averaging the numbers within the period. This is because in real applications once the

345 diesel engine is started, it should remain in service for a minimum amount of time of at least

346   one hour [26, 27] and short-term intermittence should be met by power quality improvement

347   technologies such as rechargeable batteries.

348   Figure 3 shows a one-year electricity consumption profile of the isolated village on a day-to-

349   day basis starting from the first of January. This reveals that the power consumption in winter

350   is higher than in summer due to space heating. Figure 4 indicates accordingly, the electricity

351   requirement distribution based on a 2-hour interval. It is found that the power requirement was

352   mainly in the region of 20kW to 80kW. However, the maximum electricity requirement is as

353   high as 180kW. As a result the high capacity generation has to be installed to supply the peak

354   demand, but will ultimately be idle most of the time. It should be noted in Figure 3, the

355   average daily electricity requirement shows the peak value is lower than those based on a 2-

356   hour interval in Figure 4.

357

358   The conventional generation capacities of two different schemes with diesel-only engines are

359   used as benchmarks to evaluate and compare the performance of the diesel-CAES integrated

360   system. Three systems considered in this study are:

361   • System 1: This system is a non-charged diesel engine. In order to produce electricity

362   for the end-users independently, the rated capacity of the engine is set to equal the

363   maximum load requirement for a full year $W_{DE}^{\max} = W_{EU}^{\max}$.

364   • System 2: This system is made up of two non-charged diesel engines. One is used as

365   the base capacity while the other is used as the peak load. The rated capacity of the

366   base load engine is equal to the average load of the end-users $W_{DE1}^{\max} = \overline{W}_{EU}$ where $\overline{W}_{EU}$ is

367   the annual average load of the end-users. And the peak load engine helps cover the

368   maximum load requirement with the rated capacity of $W_{DE2}^{\max} = W_{EU}^{\max} - \overline{W}_{EU}$ (generally

369   $W_{DE2}^{\max} > W_{DE1}^{\max}$). This system operates based on the following rules: in the case that the

370   end-users' load is lower than the average load, only the base load engine works. Else

371   when the end-users' load is lower than the peak engine's rated capacity, only the peak

372   engine works. Otherwise while the end-users' load is higher than the peak engine's

373   rated capacity, both the engines are turned on to accommodate the end-users' demand.

374   It should be noted that although theoretically more diesel engines with different

375   capacity can be adopted, it causes much more frequent starting and stopping which has

376   negative effects on the efficiency and lifetime of the engine [3].

377   • System 3: This is the diesel-CAES integrated system. The maximum electricity output

378   of this system with supercharging is set as $W_{SD}^{\max} = \phi \cdot W_{EU}^{\max}$ where $\phi = 1.05$ is the safety

379   coefficient. This is because the maximum power output of the integrated system

380   depends on the storage pressure of the compressed air. If the storage pressure P_S is

381   lower than the reference pressure of air turbine, the maximum power output of the

382   integrated system is then lower than the rated value. Apart from the scale of the engine

383   and the settings described in Section 3, the other key parameters for the integrated

384   system are listed in Table 3.

385

386   The fuel consumption rate and efficiency (diesel engine efficiency for system 3) of the three

387   systems are shown respectively in Figures 5 and 6 (in order to make the images clearer, only

388   one point is plotted per day). It demonstrates that the idling fuel consumption plays an

389   important role in the performance of the diesel engine. System 1 has the biggest scale diesel

390   engine and, as a result, the largest idling fuel consumption. This leads to a fuel consumption

391   rate of 22 L/hour or higher even at idling or low load factor operation, resulting in the engine

392   operating at an efficiency that is lower than 20% for the majority of the time. As shown in the

393   figures, Systems 2 using two engines as an alternative is an efficient way to save fuel. As

394 shown in Figure 5 the two diesel engines in System 2 operate simultaneously while the

395 electricity demand is higher than the single engine's rated power. During these times, fuel

396 consumption and efficiency are almost the same as System 1. However, when the end-users'

397 demand is lower than the single engine's rated power, only one engine is used. Consequently

398 the fuel rate decreases lower than 15 L/hour, enabling efficiency to be maintained in excess of

399 20%. This advantage avoids the diesel engine operating at very low load factors and, as a

400 result, significantly improves the overall performance of the system.

401

402 Compared with System 2, System 3 enhances performance by removing the idling fuel

403 consumption at lower demands. As seen in Figure 5, the diesel engine is turned off

404 (operational mode 5) for a considerable part of the operating times when the end-users'

405 demand is powered solely by the air turbine. This keeps the diesel engine operating at only

406 high load factors and, as a result, the efficiency is always more than 25%. Figure 6 illustrates

407 the high efficiency of the engine in System 3, which is often as high as 53% when the diesel

408 engine is supercharged. It should be noted that this efficiency, as defined in Section 3, is

409 higher than the traditional thermal efficiency. This is becasue the diesel engine is externally

410 powered by the air turbine. The overall fuel consumption of the diesel-CAES integrated

411 system is far less than Systems 1 and 2, as shown in Figure 7. This results in the annual diesel

412 fuel consumption of the three systems being $242m^3$, $158m^3$ and $121 \ m^3$, respectively. Thus,

413 using System 2 to replace System 1 leads to a reduction in fuel of 34.7%. Moreover, using

414 system 3 to replace system 2 brings a further 23.4% reduction, thanks to the integration of the

415 compressed air energy storage unit.

416

417 Overall, one can see that system 1 has only one operational mode, while system 2 and system

418 3 have 3 and 5 operational modes respectively. The more operational modes the system has,

419   the higher the efficiency it works at to keep variable load profiles consistent. Figure 8 plots the

420   operational modes of system 3 at varying times. Apart from the most frequent operational

421   modes 5 and 2, operational modes 3 and 4 account for a large part of the overall operation.

422   The participation of operational modes 3 and 4 changes the distribution pattern of the

423   electricity generated by the diesel engine, as illustrated in Figure 9. When comparing with

424   Figure 4, the load of the diesel engine below 50 kW is removed, partially by operational

425   modes 3 and 4 to a region higher than 100 kW. This indicates that the piston compressor is an

426   additional help to the diesel engine when operating at very high load factors and, as a result,

427   enhances the overall performance of the system.

428

429   The overall performance of the CAES unit is important to the system. Because of the energy

430   losses in the compression and expansion processes, the isolated CAES (without combustion) is

431   restricted due to its low efficiency. However, in the diesel-CAES integrated system this

432   disadvantage is overcomed by the recovery of the waste heat in the flue gas. Figure 10 shows

433   the overall energy generation of the diesel engine in three systems. It is found the net power

434   generation of the diesel engine in system 3 is 408 MWh, which is lower than those of system 1

435   and 2 (419 MWh). Therefore, the overall power generation of the air turbine is greater than the

436   overall power consumption of the piston compressor. This results in an effective energy

437   storage efficiency (the ratio of energy generated by air turbine and energy consumed by piston

438   compressor) of more than 100% due to the use of heat recovery. This is easily understandable

439   due to the high temperature of the flue gas being 200°C to 500°C or even higher. From the

440   thermodynamic view, heating the compressed gas from ambient to 300 °C prior to expansion

441   roughly doubles the net power output. In particular, as the air expansion is an open cycle the

442   compressed air can be heated up as high as possible even with variable flue gas, making the

443   heat recovery process quite efficient.

444

445   The advanced pressure storage vessel is always an issue for small scale CAES as it is costly to

446   develop and to safety-test. Therefore a smaller volume of the advanced pressure vessel makes

447   the diesel-CAES integrated system more competitive from an economic aspect. In the above

448   example, the volume of the vessel is set at $10m^3$ with the real-time storage pressure illustrated

449   in Figure 11. Increasing the vessel volume will decrease the pressure fluctuation as shown in

450   Figure 12, while based on the simulation changes of the fuel consumption are negligible (the

451   annual average efficiency increases from 34.7% to 34.9% when changing the storage volume

452   from $10m^3$ to $20m^3$). However by decreasing the storage volume to a lower than critical value

453   this may cause failure of the electricity supply as the peak load generation (operational mode 1)

454   requires continuous compressed air supply. The time interval is the key parameter affecting

455   the critical storage volume. Generally the decrease in the time interval results in a proportional

456   decrease of the critical storage volume. For example the critical storage volumes for system 3

457   above are respectively $10m^3$, $4.75m^3$ and $2.8m^3$ for the time intervals of 2 hours, 1 hour and

458   0.5 hour. However, it should be noted that in the above simulation the switching time of

459   different operational modes are ignored, while in practice the heating up of compressed air

460   may take several minutes or even longer. Furthermore, the frequent switch of the operational

461   mode also shortens its lifetime as mentioned. As a result, selection of the storage volume is a

462   balance of contributing factors mentioned above, together with the costs of power quality

463   improvement and load following facilities.

464

465   As discussed above, the diesel-CAES integrated system in this study is proposed to efficiently

466   generate electricity for isolated end-users with highly variable load patterns. Therefore the

467   characteristics of the load pattern play a pivotal role in assessing the performance of the

468   system. The load patterns of the end-users vary significantly with the location, the users'

469 composition, economics and many other factors. However in this first step of a feasibility

470 study a statistical parameter named 'Normalized standard deviation' is used to roughly

471 evaluate the characteristics of the load profile defining as:

472 $$\sigma = \sqrt{\frac{1}{N}\sum_{t=1}^{N}\left(\frac{W_{EU}^{t}}{W_{EU}^{max}} - \frac{\overline{W}_{EU}}{W_{EU}^{max}}\right)^2}$$

(Eq. 12)

473 In equation 12 W_{EU}^{t} is the end-users' load in a specific time (in this study the average load is

474 within 2 hours), $N = 4380$ is the number of timed intervals in a full year. From this

475 definition one can see the specific load is normalized first by dividing the max load in the year

476 W_{EU}^{max} and then the standard deviation is calculated in a normal way. As a result the

477 normalized standard deviation measures the normalized spread of the specific load when

478 calculating the yearly mean load.

479

480 Using the example load profile in Figure 3 as the baseline, different load profiles are

481 developed numerically by changing the altitude and the base-load. If these developed loads are

482 supplied independently by the three systems discussed above, the annual average efficiencies

483 are plotted in Figure 13. It is found the efficiency of system 1 decreases linearly with the

484 increase of the normalized standard deviation, indicating the single engine is not suitable for a

485 highly variable load. In comparison, system 2 can efficiently cover slightly changing load

486 profiles with the normalized standard deviation lower than 0.14, otherwise the annual average

487 efficiency decreases linearly as system 1. For system 3 the result is adverse. When the load is

488 very stable with the normalized standard deviation lower than 0.14, the annual average

489 efficiency decreases while the normalized standard deviation goes down. Alternatively, the

490 annual average efficiency keeps a high level if the normalized standard deviation is higher

491 than 0.14. This again suggests that system 3 has a greater advantage, particularly for highly

492 variable end-users, in particular for the cases with high share of wind power generation. Of

493 course it should be noted that the efficiency of system 3 is always much higher than system 1

494 and system 2, in part thanks to the efficient recovery of the waste heat in the flue gas.

495

496 **5. Conclusions and future work**

497 This paper proposes a diesel-CAES integrated system to supply electricity for isolated end-

498 users such as a remote village. The integrated system has five operational modes with the

499 operational principles developed accordingly. Using a single diesel system and a dual-diesel

500 system as baselines, the system performance is numerically studied to power a UK small-scale

501 village solely. The results show the fuel consumption of the integrated system is only 50% of

502 the single diesel system and 77% of the dual-diesel system. Meanwhile, the volume of the

503 high-pressure vessel for such an integrated system is found to be feasible approximately $5m^3$

504 for use in a small village with an interval time of 1 hour. The characteristics of the end-users'

505 load pattern is also studied using a statistical parameter named 'Normalized standard deviation'

506 and shows the integrated system performs very well, particularly for highly variable load

507 patterns or for high share wind power generation. The authors are currently working on the

508 construction of a lab-scale pilot system and the results will be reported in the near future.

509

514

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