

Original citation:

Feasibility study of a hybrid wind turbine system - integration with compressed air energy storage. IEEE Transactions on Energy Conversion, 32 (1). pp. 137-145.
doi:[10.1109/TEC.2016.2594285](https://doi.org/10.1109/TEC.2016.2594285).

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Feasibility Study of a Hybrid Wind Turbine System - Integration with Compressed Air Energy Storage

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Abstract—Wind has been recognized as one of major realistic clean energy sources for power generation to meet the continuously increased energy demand and to achieve the carbon emission reduction targets. However, the utilisation of wind energy encounters an inevitable challenge resulting from the nature of wind intermittency. To address this, the paper presents the recent research work at Warwick on the feasibility study of a new hybrid system by integrating a wind turbine with compressed air energy storage. A mechanical transmission mechanism is designed and implemented for power integration within the hybrid system. A scroll expander is adopted to serve as an “air-machinery energy converter”, which can transmit additional driving power generalized from the stored compressed air to the turbine shaft for smoothing the wind power fluctuation. A mathematical model for the complete hybrid process is developed and the control strategy is investigated for corresponding cooperative operations. A prototype test rig for implementing the proposed mechanism is built for proof of the concept. From the simulated and experimental studies, the energy conversion efficiency analysis is conducted while the system experiences different operation conditions and modes. It is proved that the proposed hybrid wind turbine system structure is feasible technically.

Index— wind turbines, compressed air energy storage, hybrid systems, mathematical modelling, control strategy.

1. Introduction

In recent years, wind power generation has shown a robust growth trend worldwide. The global cumulatively installed generation capacity of wind power reached 318,137 MW at the end of 2013, which has increased by more than 163% compared to 120,624 MW in 2008 [1]. Such rapid development is mainly driven by the continuous increase in electricity demand and the need for reducing greenhouse gas emissions. However, the nature of fluctuation and intermittence of wind makes it very difficult to deliver power output from wind energy with an instant match to the electricity demand. This nature also brings the negative impact

33 onto the wind turbine system operation efficiency, life expectancy and mechanical structures [2]. Thus, new
34 technologies and approaches have been actively researched to alleviate the problems caused by wind
35 fluctuation and intermittence, such as wind turbine pitch angle control, power electronics development for
36 wind power and flexible back-up power generation [3, 4, 5]. One of the promising solutions is to introduce an
37 element of stored energy as an alternative energy supply for use when the ambient wind power is insufficient.
38 Various Energy Storage (ES) technologies can provide the service of compensators to work with different
39 types of wind power generation systems, for example, hydroelectric pumped storage, Compressed Air Energy
40 Storage (CAES), flow batteries and flywheels [6, 7]. Among the available ES technologies, CAES can be
41 considered as one of the relatively mature and affordable options [6, 8, 10].

42 CAES technology refers to storing energy in the form of high pressure compressed air during the periods of
43 low electrical energy demand and then releasing the stored energy during the high demand periods. CAES
44 facilities exist in multiple scales, with long storage duration, moderate response time and good part-load
45 performance [6, 7, 9]. So far, there are a few successful industrial implementations of large-scale CAES plants
46 serving wind power generation. For instance, after the world first commercialized Huntorf CAES plant started
47 operation, its mandate was updated to include the buffering against the intermittence of wind energy
48 production in Northern Germany [9]. Also, the developing advanced adiabatic CAES demonstration project –
49 ADELE by RWE Power and others aims to store large amounts of electrical energy through CAES and
50 thermal storage concepts; the ADELE plans to operate with wind farms, with a storage capacity of 360 MWh
51 without CO₂ emissions [11, 12].

52 In addition to the large-scale CAES facility integrated with the wind power generation, the work presented
53 in the paper is to explore the potential of using smaller scale CAES in the wind power application. Inspired
54 by the parallel drive train in Hybrid Electrical Vehicles (HEVs) ([13]), this paper presents a novel direct
55 electromechanical integration of a wind turbine system and a CAES mechanism at a few kW scale. The
56 objective is to develop a system with simple structure, efficient, low maintenance, clean and sustainable. The
57 proposed design is illustrated in Fig. 1. It consists of three main sections:

58 (1) Wind turbine subsystem: this subsystem simulates a real scenario of horizontal wind turbines' operation. It
59 includes a module of wind power extracted by blades, a mechanical drive train, a Permanent Magnet
60 Synchronous Generator (PMSG) and its load(s) to be driven. The generated electricity can be directly used
61 to end-users or fed back to grids via electric power converters and inverters.

62 (2) CAES subsystem: it is composed of a scroll expander and a compressed air storage tank. This relatively
 63 new type of expander has a smart mechanical structure leading to a higher energy conversion ability
 64 compared to most other pneumatic drives. Due to the capacity of typical scroll expanders, the proposed
 65 structure is more suitable for small-scale wind turbine systems. The compressed air stored in the tank can
 66 be obtained from the operation of compressors on site or local suppliers. From Fig. 1, through a mechanical
 67 transmission mechanism, an additional driving power by the CAES subsystem can provide a direct
 68 compensation to the wind turbine.

69 (3) Controller: for managing the whole hybrid system's operation, an appropriate control strategy is
 70 particularly important for supporting the system multi-mode operations and ensuring the dynamic balance
 71 of driving power and electric load demand.

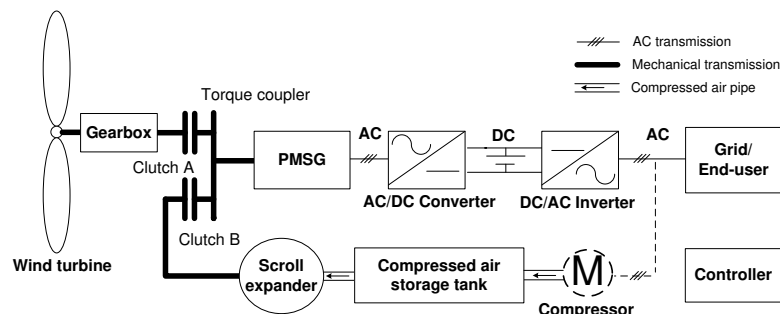


Fig.1 Small-scale hybrid wind turbine with CAES

72
73

74 Study of hybridization of wind generation with CAES was reported in various literatures, for example, [14-
 75 17]. The common feature of the previously reported hybridization systems is that CAES is treated as an
 76 independent energy storage unit and is engaged with wind power generation through management of
 77 electricity network connection. The hybrid system proposed in this paper is mainly new and different because
 78 the CAES is directly connected to the turbine shaft through a mechanical transmission mechanism. In this way,
 79 with a proper control strategy, the compressed air energy will be released via the direct mechanical connection
 80 to contribute to wind turbine power generation. Thus the system does not require a separate generator and
 81 extra electricity conversion device(s) which will reduce the whole system cost. In addition, the extra torque
 82 input from the air expander could reduce the turbine shaft stress for prolonging the turbine life time.

83 The paper starts from description of the hybrid system, development of its mathematical model, and
 84 presentation of a suitable multi-mode control strategy. Then a hybrid wind turbine test rig is reported, which is
 85 installed in the authors' research laboratory. Finally the whole system energy conversion efficiency analysis is
 86 given.

87 **2. Mathematical model of the hybrid wind turbine system**

88 In this subsection, the mathematical models for a typical wind turbine, a Permanent Magnet Synchronous
 89 Generator (PMSG), a scroll expander and a novel mechanical power transmission system are presented, and
 90 then the whole system control strategy is described. In the modelling study, it is assumed that the air supply of
 91 the scroll expander, i.e., the compressed air from the storage tank, is sufficiently pre-compressed air with
 92 constant temperature. Thus the scroll expander air supply can be regarded as a controllable compressed air
 93 source.

94 **2.1 Mathematical model for wind turbines**

95 A typical horizontal axis wind turbine is chosen in the hybrid system for modelling study. Its mechanical
 96 power output P which can be produced by the turbine at the steady state is given by:

97
$$P = \frac{1}{2} \rho_a \pi r_T^2 v_w^3 C_p \tag{1}$$

98 where ρ_a is the air density; r_T is the blade radius; v_w is the wind speed; C_p represents the turbine efficiency,
 99 revealing the capability of turbine for obtaining energy from the wind. This coefficient depends on the tip
 100 speed ratio and the blade angle. Because the calculation of C_p requires the knowledge of aerodynamics and
 101 the computations are quite complicated, some numerical approximations to (1) were developed and studied
 102 [18, 19]. In this hybrid system modelling, the following function is adopted to approximate the calculation
 103 presented in (1) [18, 20],

104
$$C_p(\lambda, \theta) = 0.22 \left[\frac{116(\lambda\theta^3 + \lambda + 0.08\theta^4 + 0.08\theta)}{\theta^3 - 0.035\lambda - 0.0028\theta + 1} - 0.4\theta - 5 \right] e^{-\frac{12.5(\lambda\theta^3 + \lambda + 0.08\theta^4 + 0.08\theta)}{\theta^3 - 0.035\lambda - 0.0028\theta + 1}} \tag{2}$$

105 where θ represents the pitch angle, λ stand for the tip speed ratio, $\lambda = \omega_T \cdot r_T / v_w$, ω_T is the turbine speed. Eq.
 106 (2) lead to $C_p(\lambda, \theta)$ versus λ characteristics for various values of θ as depicted in Fig. 2. It can be seen that the
 107 power coefficient C_p varies with different values of the pitch angle θ (for instance, $\theta = 0^\circ, 5^\circ, 10^\circ$ and 15° as
 108 shown in Fig. 2), and the best efficiency is obtained at $\theta = 0^\circ$ in most cases [21]. From the above, the
 109 mechanical driving power extracted from the wind can be calculated by Eq. (1) and (2).

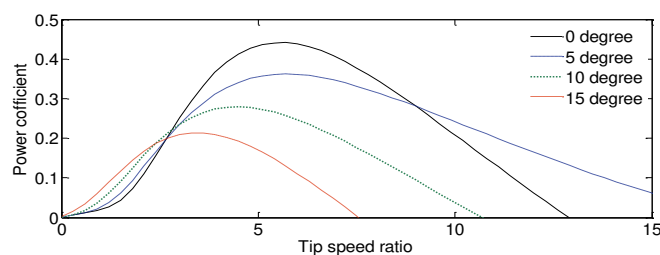


Fig. 2 Power coefficients as a function of tip speed ratio and pitch angle

110
111

112 The drive train of a wind turbine system normally consists of a blade pitching mechanism with a spinner, a
 113 hub with blades, a rotor shaft, a gearbox with brake and sometimes a generator. The generator impact on the
 114 whole hybrid system will be considered in the mechanical power transmission modelling in the later
 115 subsection. Thus to the proposed system, for describing the dynamic behaviours of the pure wind turbine, a
 116 simplified mathematical model is considered,

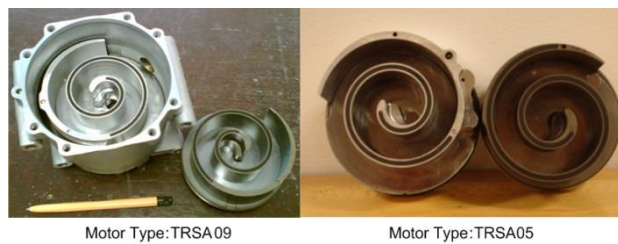
$$117 \quad \frac{d}{dt} \omega_T = \frac{1}{J_T} (\tau_T - \tau_L - B\omega_T)$$

118 (3)

119 where ω_T and J_T are the rotation speed and the inertia of turbine blades respectively, τ_T and τ_L stand for
 120 the torques of wind turbine and low-speed shaft individually, B is the damping coefficient of the driven train
 121 system. The low-speed shaft is connected and driven by the turbine rotor.

122 2.2 Mathematical model for scroll expanders

123 The scroll expander, also known as the scroll type air motor, is a relatively new member to pneumatic
 124 drives. Such type of expander is famous with its high efficiency and its unique smart mathematical structure
 125 [22, 23], which is the key component in the proposed small-scale hybrid wind turbine system. Fig. 3 shows the
 126 mechanical structure of a typical scroll expander. It can be seen that, inside the expander shell, there are two
 127 intermeshed identical scrolls, namely the moving scroll and the fixed scroll. Each scroll is fitted with a back
 128 plate. Both two scrolls are circular involutes. One scroll is mirrored with respect to the other. The crank shaft
 129 of the scroll expander connects to the back plate of the moving scroll through a cam and bearing mechanism.



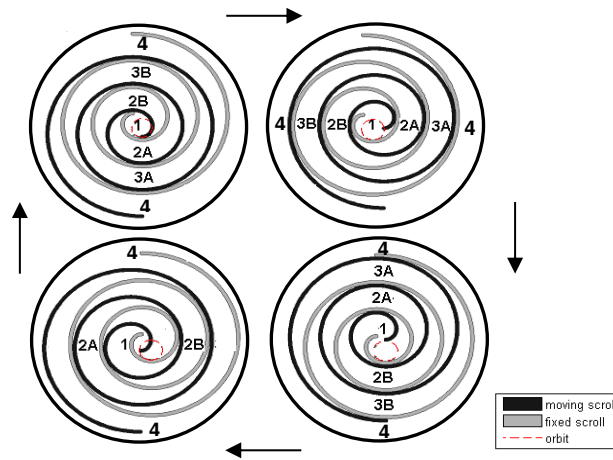
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131

Fig. 3 Illustration of the scroll expander structure (manufactured by Sanden)

132 A scroll expander with three wraps in motion and its moving scroll orbit trajectory is illustrated in Fig. 4.
 133 The black scroll stands for the moving scroll and the grey one represents the fixed scroll. The moving scroll
 134 travels along the orbit anticlockwise when the compressed air comes into the scroll mechanism. During the
 135 expander operation, these two scrolls always keep contacting at some points. This forms three different types
 136 of air chambers inside the expander shell: a central chamber, even number of sealed crescent chambers and an
 137 exhaust chamber. The early work by the authors has proven that the scroll expander has more energy efficient

138 performance compared to conventional pneumatic drives with similar scales (up to several kW level), such as
 139 reciprocating cylinders, vane type air motors, etc. [24].



140
 141

Fig. 4 Schematic diagram of a scroll expander in motion

142 With the following assumptions: 1) no air leakage, 2) the scroll expander using ideal air and 3) it working at
 143 a constant temperature environment, a simplified mathematical model for scroll expanders can be derived [22,
 144 24, 25]. The geometric model for scroll expanders can be derived from the fundamental curve of a spiral. The
 145 equations for the moving scroll can be described by,

$$146 \quad x_A(\varphi_s, \alpha_s) = x_0 + (\rho_0 + \kappa_s \varphi_s) \sin \varphi_s + \kappa_s \cos \varphi_s - \kappa_s + r_s \sin \alpha_s$$

147 (4)

$$148 \quad y_A(\varphi_s, \alpha_s) = y_0 - (\rho_0 + \kappa_s \varphi_s) \cos \varphi_s + \kappa_s \sin \varphi_s + \rho_0 - r_s \cos \alpha_s$$

149 (5)

150 where (x_0, y_0) is the initial position and ρ_0 is the initial curvature radius for the moving scroll curve, κ_s is
 151 the slope of the curvature radius, r_s refers to the orbit radius of the moving scroll, α_s stands for the scroll
 152 expander orbit angle, φ_s is the tangential angle to the moving scroll. The fixed scroll is generated by the curve
 153 which envelops the family of the moving scroll curves when the moving scroll wobbles along with its orbit
 154 [22]. The equations for the fixed scroll can be,

$$155 \quad x_B(\phi_s) = x_1 - (\rho_0 + \kappa_s \phi_s) \sin \phi_s - \kappa_s \cos \phi_s + \kappa_s \quad (6)$$

$$156 \quad y_B(\phi_s) = y_1 + (\rho_0 + \kappa_s \phi_s) \cos \phi_s - \kappa_s \sin \phi_s - \rho_0 \quad (7)$$

157 where ϕ_s is the tangential angle to the fixed scroll, (x_1, y_1) is the initial position for the fixed scroll curve, and
 158 the moving scroll contacts the fixed scroll at the points, thus $\varphi_s = \phi_s + j\pi$, j is an arbitrary integer [22].

159 Applying Green's Theorem, the equations for describing the volume variations of the scroll expander
 160 chambers can be derived [25]. The control volume of the central chamber is,

$$161 \quad V_c(\alpha_s) = z \left[(x_0 \kappa_s \pi - \kappa_s^2 \pi - x_0 r_s + r_s \kappa_s) \cos \alpha_s + \kappa_s^2 \pi \alpha_s^2 + (r_s \rho_0 \kappa_s \pi - r_s \rho_0 - y_0 r_s + y_0 \kappa_s \pi) \sin \alpha_s - r_s \kappa_s \right. \\ \left. + \frac{1}{3} \kappa_s^2 \pi^3 + (r_s \pi \kappa_s + 2 \kappa_s \rho_0 \pi) \alpha_s - \frac{1}{2} r_s \pi^2 \kappa_s + \rho_0 r_s \pi + \frac{1}{2} r_s^2 \pi + \rho_0^2 \pi \right] \quad (8)$$

162 where $V_c(\alpha_s)$ is the volume of scroll expander central chamber, z is the depth of the moving and fixed scrolls.

163 The control volume of the i^{th} ($i=1, 2, 3 \dots$) pair of sealed crescent chambers is:

$$164 \quad V_s(\alpha_s, i) = z [\pi r_s^2 + 2 \pi r_s (\rho_0 + \kappa_s (\alpha_s + \pi + 2(i-1)\pi))] \quad (9)$$

165 where $V_s(\alpha_s, i)$ is the volume of scroll expander sealed crescent chamber volume. The control volume of the
 166 exhaust chamber can be described by,

$$167 \quad V_e(\alpha_s) = V_{total} - V_c(\alpha_s) - \sum V_s(\alpha_s, i) \quad (10)$$

168 where $V_e(\alpha_s)$ is the volume of scroll expander exhaust chamber volume, V_{total} represents the total control
 169 volume of the scroll expander.

170 From the fundamental of thermodynamics and the theory of orifice, the air pressure of the different scroll
 171 expander chambers can be calculated [22, 24, 25]. To the air pressure variation of the central chamber (\dot{p}_c),

$$172 \quad \dot{p}_c = -\frac{\dot{V}_c}{V_c} p_c \omega_s \gamma + \frac{1}{V_c} \gamma \mathcal{R} C_d C_0 C_k A_s p_s f(p_c / p_s) \sqrt{T_s} \quad (11)$$

173 To the air pressure variation of the first pair of sealed crescent chambers (\dot{p}_{s1}),

$$174 \quad \dot{p}_{s1} = -\frac{\dot{V}_s(\alpha_s, 1)}{V_s(\alpha_s, 1)} p_{s1} \omega_s \gamma \quad (12)$$

175 To the air pressure variation of the second pair of sealed crescent chambers (\dot{p}_{s2}),

$$176 \quad \dot{p}_{s2} = -\frac{\dot{V}_s(\alpha_s, 2)}{V_s(\alpha_s, 2)} p_{s2} \omega_s \gamma \quad \alpha_s \in [0, \pi] \quad (13)$$

177 To the air pressure variation of the exhaust chamber (\dot{p}_e),

$$178 \quad \dot{p}_e = -\frac{\dot{V}_e}{V_e} p_{s2} \omega_s \gamma + \frac{1}{V_e} \gamma \mathcal{R} C_d C_0 C_k A_o p_e f(p_{atm} / p_e) \sqrt{T_s} \quad (14)$$

179 The driving torque generated by a scroll expander is the sum of torques on all driving segments on the two
 180 scrolls, and it can be derived as [22, 24, 25],

$$181 \quad \tau_s = \begin{cases} z r [(2 \rho_0 + 2 \kappa_s \alpha_s + \kappa_s \pi)(p_c - p_{s1}) + (2 \rho_0 + 2 \kappa_s \alpha_s + 5 \kappa_s \pi)(p_{s1} - p_{s2}) + (2 \rho_0 + 2 \kappa_s \alpha_s + 9 \kappa_s \pi)(p_{s2} - p_e)] & \alpha_s \in [0, \pi] \\ z r [(2 \rho_0 + 2 \kappa_s \alpha_s + \kappa_s \pi)(p_c - p_{s1}) + (2 \rho_0 + 2 \kappa_s \alpha_s + 5 \kappa_s \pi)(p_{s1} - p_e)] & \alpha_s \in (\pi, 2\pi] \end{cases} \quad (15)$$

182

183 where p_s is the supply pressure, p_{atm} is the pressure of atmosphere, T_s is the supply temperature, R is the gas
184 constant, $c_0 = 0.04$, $c_d = 0.8$, $c_k = 3.864$, $\gamma = 1.4$ is the ratio of specific heat, A_i , A_o are the effective area
185 of expander inlet and outlet respectively, r_s is the radius of the orbit, ω_s is the rotation speed of scroll
186 expander shaft, $f()$ is a function of the ratio between the downstream and upstream pressures at the orifice
187 [24, 25]. In the modelling, it should be noticed that, when the orbit angle $\alpha_s \in (\pi, 2\pi]$, the second pair of
188 crescent chambers is not sealed anymore in each period (refer to Fig. 4).

189 2.3 Mathematical model for permanent magnet synchronous generators

190 A Permanent Magnet Synchronous Generator (PMSG) has been chosen as the driven machine of the wind
191 turbine; a resistive load is directly connected to the PMSG electricity output for simplicity of analysis. The
192 mathematical model is described by Eq. (16-22), which has been studied in [26-28]:

$$193 \quad \frac{d\omega_G}{dt} = \frac{1}{J_G}(\tau_G - \tau_e - F_G\omega_G) \quad (16)$$

$$194 \quad \frac{d\theta_G}{dt} = \omega_G \quad (17)$$

$$195 \quad \frac{di_d}{dt} = \frac{1}{L_d}v_d - \frac{R_G}{L_d}i_d + \frac{L_q}{L_d}p_G\omega_G i_q \quad (18)$$

$$196 \quad \frac{di_q}{dt} = \frac{1}{L_q}v_q - \frac{R_G}{L_q}i_q - \frac{L_d}{L_q}p_G\omega_G i_d - \frac{\varepsilon p_G \omega_G}{L_q} \quad (19)$$

$$197 \quad \tau_e = 1.5p_G[\varepsilon i_q + (L_d - L_q)i_d i_q] \quad (20)$$

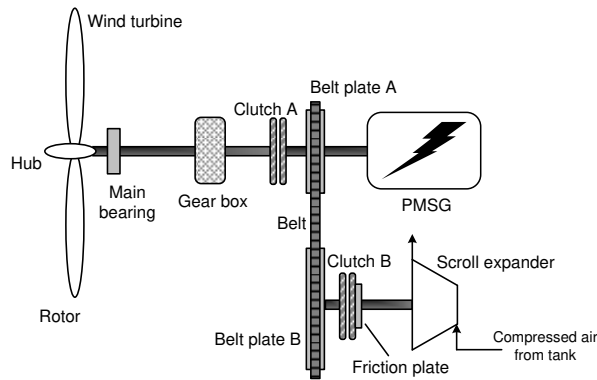
$$198 \quad v_q = \frac{1}{3}[\sin(p_G\theta_G) \cdot (2v_{ab} + v_{bc}) + \sqrt{3}v_{bc} \cos(p_G\theta_G)] \quad (21)$$

$$199 \quad v_d = \frac{1}{3}[\cos(p_G\theta_G) \cdot (-2v_{ab} - v_{bc}) - \sqrt{3}v_{bc} \sin(p_G\theta_G)] \quad (22)$$

200 where the subscripts a, b, c, d, q mean the a, b, c, d, q axis respectively, θ_G and ω_G are the PMSG rotor angular
201 position and speed respectively, τ_G and τ_e stand for the PMSG driving and electromagnetic torques, J_G is the
202 inertia of the PMSG, R_G is the resistance of the stator windings, L_q , L_d are the resulted q and d axis
203 inductances respectively, p_G is the number of PMSG pole pairs, i and v are the current and voltage in the
204 different axes, ε is the flux amplitude induced by the permanent magnets of the rotor, F_G is combined viscous
205 friction of the generator rotor. The Park's transformation is employed for transforming \vec{X}_{abc} (3-phase
206 coordinates) to \vec{X}_{dq} (DQ rotating coordinates) [26, 28].

207 2.4 Mathematical model for the mechanical power transmission

208 The designed power transmission system mainly includes two electromagnetic clutches and a belt speed
 209 transmission to ensure coaxial running, as shown in Fig. 5. The functions of two clutches are described below:
 210 (1) Clutch A is engaged in almost all cases. Unless the wind speed is extremely low - under the cut-in wind
 211 speed, Clutch A will be disengaged and then the PMSG will be exclusively driven by the scroll expander. For
 212 the simplicity of modelling, the extreme low wind speed (Clutch A disengagement) situation is not considered.
 213 (2) Clutch B is placed to the hybrid system for conditional switching on/off the small-scale CAES subsystem.
 214 When the wind turbine cannot generate sufficient electricity to match the electric load demand, the compressed
 215 air in the storage tank will be released into the scroll expander via a pneumatic valve and/or regulator's control;
 216 then the scroll expander will start rotating and Clutch B will be engaged at the moment of the expander rotor
 217 speed comparably to the wind turbine shaft speed after the belt transmission. Also, Belt plate A and B of the
 218 belt transmission have different diameters to play the function as a gearbox (refer to Fig. 5). Thus the small-
 219 scale CAES subsystem and the wind turbine can be integrated rigidly.



220
 221 Fig. 5 Structure of the mechanical power transmission in the hybrid wind turbine system

222 According to the above description, the main working states with their mathematical expressions of the
 223 mechanical power transmission can be derived:

224 **Case I.** Clutch A engaged and Clutch B disengaged: the two disks of Clutch B is fully separated (refer to Fig.
 225 5). Considering friction and different payloads, applying Newton's second law of angular motion, to the shaft
 226 of scroll expander, we have,

$$227 \quad \tau_s - M_f \omega_s = (J_s + J_f) \dot{\omega}_s \quad (23)$$

228 where J_s is the scroll expander inertia, J_f is the friction plate inertia, τ_s is the scroll expander driving torque;
 229 M_f is the combined viscous friction coefficient; $\dot{\omega}_s$ represents the scroll expander angular acceleration.

230 To the main shaft of the wind turbine system, both the active plate and the passive plate of the belt
 231 transmission can be considered as an extra inertia load, thus the total equivalent inertia can be,

$$232 \quad J_{total} = J_{pass} + \zeta^2 J_{act} \quad (24)$$

233 where J_{pass} and J_{act} are the inertias of the passive and active plate respectively, ζ is the speed ratio of the
234 belt.

235 **Case II.** Both Clutch A and Clutch B engaged: Once Clutch B is engaged by coupling its two disks, the
236 following equations can be derived:

$$237 \quad \begin{cases} \tau_s - M_f \omega_s - \tau_{act} = (J_s + J_f + J_{act}) \dot{\omega}_s \\ \tau_{pass} = \tau_{act} \eta \zeta \\ \tau_H + \tau_{pass} - \tau_e - F_G \omega_G = \dot{\omega}_G (J_G + J_{pass}) \\ \omega_s = \omega_G \zeta \end{cases} \quad (25)$$

238 where τ_H is the torque of wind turbine high-speed shaft. The high-speed shaft is linked to the output of
239 gearbox (refer to Fig. 5). η is the transmission efficiency of the belt.

240 2.6 Overall state space model of the hybrid system

241 With all the subsystem models presented above, the overall state space model for the hybrid system is
242 presented below. The system state variables are chosen: x_1 : PMSG rotor angle, x_2 : PMSG angle velocity, x_3 :
243 current in d axis for PMSG, x_4 : current in q axis for PMSG, x_5 : pressure in the expander central chamber,
244 x_6 : pressure in the expander first pair of crescent chambers, x_7 : pressure in the expander second pair of
245 crescent chambers, x_8 : pressure in the expander exhaust chamber; and the input variables u_1 : pitch angle, u_2 :
246 supply pressure for the scroll expander. Integrating the wind turbine, driven train and PMSG sub-models, the
247 state functions of the wind turbine system with the engaged CAES can then be described by:

$$248 \quad \dot{x}_1 = x_2 \quad (26)$$

$$249 \quad \dot{x}_2 = \frac{1}{J_G + J_{pass} + J_T \eta_T \zeta^2 + (J_s + J_f + J_{act}) \eta \zeta^2} \left[\frac{\eta_T}{2x_2} \rho_a \pi r_T^2 v_w^3 C_p(u_1) \right. \\ \left. - B_{eq} \eta_T \zeta^2 x_2 + \eta \zeta \tau_s - M_f \eta \zeta^2 x_2 - 1.5 p_G (\epsilon x_4 + L_d x_3 x_4 - L_q x_3 x_4) - F_G x_2 \right] \quad (27)$$

$$250 \quad \dot{x}_3 = \frac{v_d}{L_d} - \frac{R_G}{L_d} x_3 + \frac{L_q}{L_d} p_G x_2 x_4 \quad (28)$$

$$251 \quad \dot{x}_4 = \frac{v_q}{L_q} - \frac{R_G}{L_q} x_4 - \frac{L_d}{L_q} p_G x_2 x_3 - \frac{\epsilon p_G x_2}{L_q} \quad (29)$$

$$252 \quad \dot{x}_5 = -\frac{\dot{V}_c}{V_c} \gamma x_5 \frac{x_2}{\zeta} + \frac{1}{V_c} \gamma R C_d C_0 C_k A_i u_2 f(x_5 / u_2) \sqrt{T_s} \quad (30)$$

$$253 \quad \dot{x}_6 = -\frac{\dot{V}_s(\alpha_s, 1)}{V_s(\alpha_s, 1)} \gamma x_6 \frac{x_2}{\zeta} \quad (31)$$

254
$$\dot{x}_7 = -\frac{\dot{V}_s(\alpha_s, 2)}{V_s(\alpha_s, 2)} \gamma x_7 \frac{x_2}{\zeta} \quad \alpha_s \in [0, \pi] \quad (32)$$

255
$$\dot{x}_8 = -\frac{\dot{V}_e}{V_e} \gamma x_8 \frac{x_6}{\zeta} + \frac{1}{V_e} \gamma R C_d C_0 C_k A_o x_8 f(p_{atm} / x_8) \sqrt{T_s} \quad (33)$$

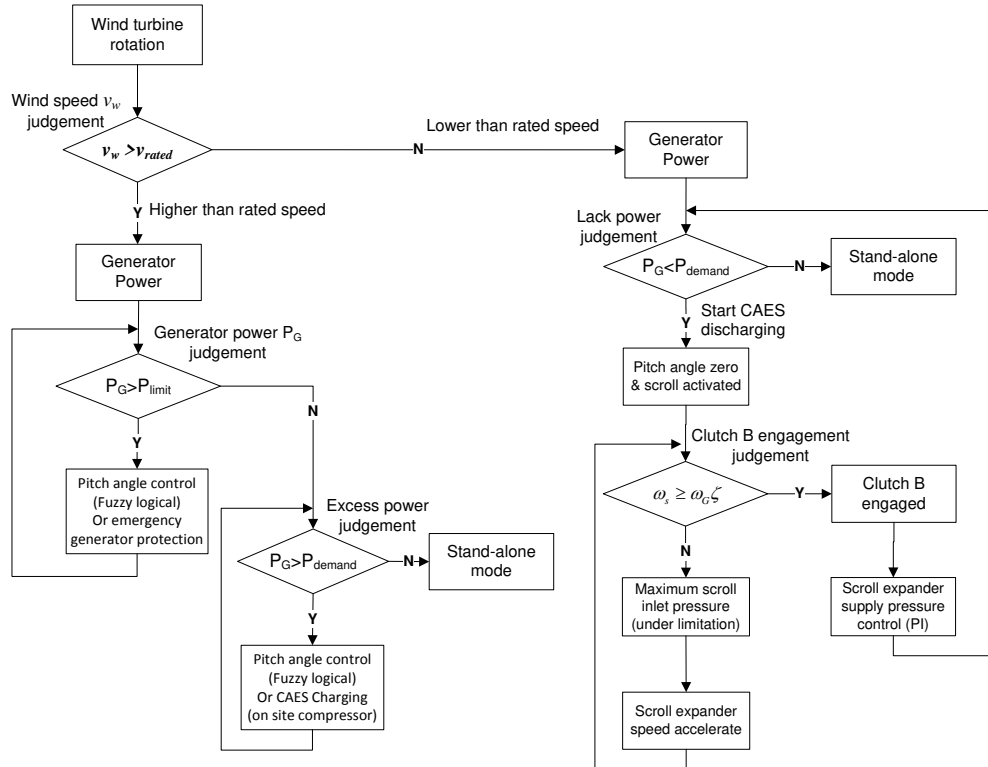
256 where η_T and ζ_T stands for the efficiency and ratio of the turbine shaft transmission, B_{eq} is the equivalent
 257 damping coefficient of wind turbine. If the CAES device is disengaged to the wind turbine system, we have:

258
$$\dot{x}_2 = \frac{1}{J_G + J_{pass} + J_T \eta_T \zeta_T^2} \left[\frac{\eta_T}{2x_2} \rho_a \pi r_T^2 v_w^3 C_p(u_1) - B_{eq} \eta_T \zeta_T^2 x_2 - 1.5 p_G (\epsilon x_4 + L_d x_3 x_4 - L_q x_3 x_4) - F_G x_2 \right] \quad (34)$$

259 with $\dot{x}_5 = \dot{x}_6 = \dot{x}_7 = \dot{x}_8 = 0$.

260 **3. Control strategy study for the hybrid wind turbine system**

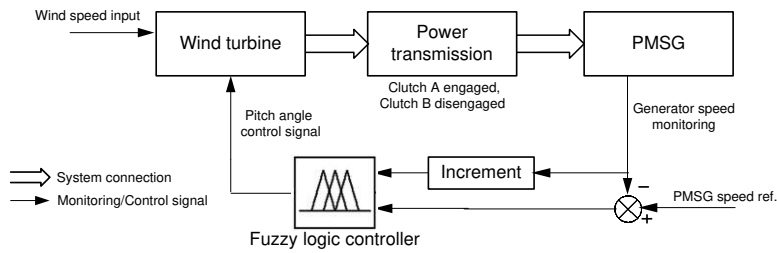
261 The whole hybrid system consists of several subsystems and has multi-mode operations. Thus it is necessary
 262 to develop a set of suitable decision-making rules to switch smoothly between different modes (e.g., stand-
 263 alone wind turbine and hybrid wind turbine integrated with CAES). It is also required to design dynamic
 264 control method(s) for the system performance optimization and load balance in each mode operation. The flow
 265 chart of the designed multi-mode control strategy for achieving the above objective is illustrated in Fig. 6. It
 266 can be seen that, for fully regulating the output power of the hybrid system to accurately match the load
 267 demand, the control strategy is required to cover all possible situations.



269

Fig. 6 Multi-mode control diagram for the proposed hybrid system

270 From the flow chart (Fig. 6), the main operation modes are introduced as follows: (1) while the PMSG
 271 output power is above its limitation ($P_G > P_{limit}$), the fuzzy logical control for pitch angle (or emergency
 272 generator protection) is adopted; (2) under the high wind speed situations, i.e, $v_w > v_{rated}$, if the PMSG output
 273 power is lower than its limitation but higher than the electric load demand, the fuzzy logical control for pitch
 274 angle can be maintained in the hybrid system (or surplus power can be used for CAES subsystem charging if
 275 an on-site compressor is available); (3) in the case of the insufficient PMSG output power cannot meet the
 276 electric load demand, CAES subsystem discharging mode is activated; when the scroll expander rotor speed
 277 reach a certain level, that is, $\omega_s \geq \omega_G \zeta$, Clutch B will be engaged and then the scroll expander output torque
 278 can be used to provide additional driving power to the wind turbine system; in this case, a PI controller is
 279 employed to regulate the supply pressure of the scroll expander and in turn to manage the power out of scroll
 280 expander; (4) when the PMSG output power roughly matches the electric load demand, the hybrid system will
 281 be running on the stand-alone mode, that is, Clutch B will be disengaged and the whole system will be in
 282 operation with no signal from the controller.



283

284

Fig. 7 Schematic of the pitch angle fuzzy logic control

285 Pitch angle adjustment is a common approach to regulate the aerodynamic power extracted by the wind
 286 turbine blade, which is the input power of the turbine system. A fuzzy logical controller is introduced to the
 287 designed control system for pitch angle adjustment to limit the power captured at the high wind speed
 288 situations (refer to Fig. 6). Fuzzy logic control provides a systematic way to incorporate human experience for
 289 controlling a nonlinear system, which is proved to be appropriate to such type of systems [29, 30]. Fig. 7
 290 shows the schematic of pitch angle fuzzy logic controller implemented in the hybrid system. Choosing the
 291 speed error V_e (difference between the PMSG actual speed and the reference speed), the PMSG actual speed
 292 increment V_i and the pitch angle control value θ_p as the linguistic variables. V_e and V_i are inputs of the
 293 fuzzy logic controller, θ_p is the controller output (refer to Fig. 7). The linguistic values of V_e and θ_p are:

294 [NB NM NS ZO PS PM PB], which means negative big, negative middle, negative small, zero, positive small,
 295 positive middle, and positive big respectively; The linguistic values of V_i are: [N Z P], which stand for
 296 negative, zero and positive. The standard triangular membership functions have been used for both the inputs
 297 and the output of the controller. The control law is represented by a set of heuristically chosen fuzzy rules
 298 which are given in Table 1.

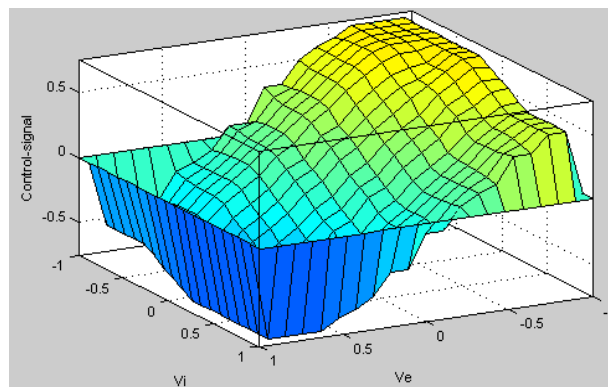
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Table 1 Rule base for proposed fuzzy controller

θ_p		V_i		
		N	Z	P
V_e	NB	PB	PB	PM
	NM	PB	PM	PS
	NS	PM	PS	ZO
	ZO	PS	ZO	NS
	PS	ZO	NS	NM
	PM	NS	NM	NB
	PB	NM	NB	NB

300

301 Based on the triangular membership functions and the fuzzy rules, the designed fuzzy logic controller can
 302 produce a crisp and continuous nonlinear input/output map as shown in Fig. 8. This map indicates that
 303 numerous nonlinearities are designed to enhance the controller's performance to drive the system to the set
 304 point. The details related to using fuzzy logic control specific to adjust the pitch angle of wind turbines can be
 305 found in [14, 29, 30].



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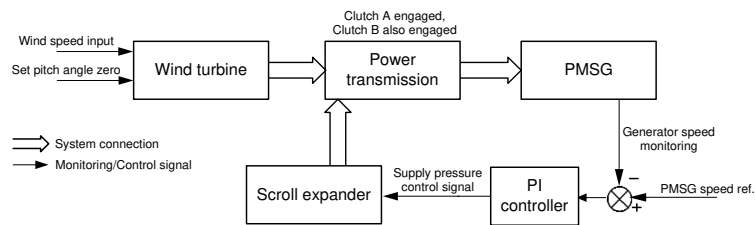
Fig. 8 Fuzzy logical controller nonlinear input & output map

308 During the low wind speed periods, the CAES device will work at the discharging mode to provide
 309 additional driving power. For controlling the input power from the CAES device to the wind turbine system, it
 310 is necessary to manage how much compressed air flows into the scroll expander at every moment. Considering
 311 the limited central chamber volume of the scroll expander, the supply air pressure control by a digital

312 proportional pressure regulator is more suitable to achieve this purpose, compared to the traditional pneumatic
 313 valve displacement control. A PI control method on the pressure regulator is chosen because of its simplicity.
 314 Fig. 9 shows the schematic of the scroll expander supply pressure PI control. The controller input is the PMSG
 315 speed tracking error $e(t)$, i.e., $e(t) = \omega_{ref} - \omega_G$, where ω_{ref} is the PMSG reference speed. The control law can
 316 be represented as:

$$317 \quad U(t) = K_p e(t) + K_I \int e(t) + C_{initial} \quad (35)$$

318 where K_p and K_I are the proportional and integral control gains, $C_{initial}$ represents the initial controller
 319 reference value. In addition, during the low wind speed periods, it is common to set the pitch angle θ equals to
 320 zero for achieving the best turbine efficiency C_p (refer to Subsection 2.1 and Fig. 9).



321
 322 Fig. 9 Schematic of the scroll expander supply pressure PI control

323 4. Simulation study for the hybrid system

324 The overall state space model of the hybrid wind turbine with the CAES system and its corresponding multi-
 325 mode control strategy are implemented in Matlab/Simulink environment for simulation study. The parameters
 326 for the simulation study of the whole hybrid system are listed in Table 2. Most parameters related to the scroll
 327 expander, the drive train and the PMSG are obtained from the associated data sheets or measurement of the
 328 machines which are used for building the experimental test rig in the laboratory. However, due to the
 329 complicated structure of the hybrid system, sometimes it is difficult to obtain the precise values for all
 330 parameters. These unknown parameters for models can be identified using intelligent computational
 331 algorithms together with the experimental data [31].

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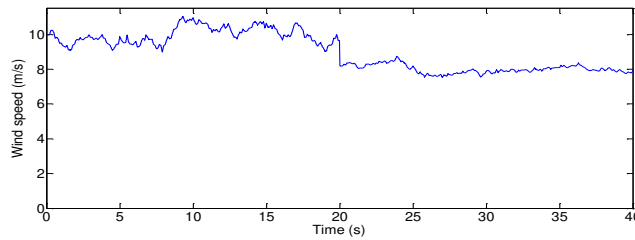
Table 2 Parameters of the hybrid wind turbine system

Symbol	Description	Value
J_T	inertia of turbine blades	$4.9 \text{ kg} \times \text{m}^2$
ρ_a	Air density	1.25 kg/m^3
r_T	Blade radius	1.75 m
ς_T	Speed ratio of turbine shaft transmission	5
η_T	Efficiency of turbine shaft transmission	0.95
η	Transmission efficiency of the belt	0.95
r_s	Orbit radius of the scroll	$5.40 \times 10^{-3} \text{ m}$
z	Depth of the scroll chambers	$3.33 \times 10^{-2} \text{ m}$

V_{total}	Total control volume of the scroll	$2.50 \times 10^{-4} \text{ m}^3$
p_G	The number of pole pairs	6
R_G	Resistance of the PMSG stator windings	1.31 Ohm
L_d	Inductance on d axis	2.075 mH
L_q	Inductance on q axis	2.075 mH
λ	Flux amplitude induced by permanent magnets	0.171Wb

334

335 The simulation considers the scenario when the input mean wind speed steps down within a 40 seconds'
336 time series observation window, as shown in Fig. 10. A white noise source with a shaping filter is chosen to
337 generate the input wind speed profile, and its feasibility had been studied in [32-34]. From Fig. 10, it can be
338 seen that the mean wind speed drops from around 10 m/s to 8 m/s at the moment of the 20th second. Thus the
339 simulated data can represent the wind speed variation in a certain period.



340

341

Fig. 10 Simulated wind speed profile

342 Introducing the simulated wind speed profile given in Fig. 10 to the hybrid system input, Fig. 11 shows the
343 comparisons of dynamic responses of the multi-mode controlled hybrid system and the stand-alone wind
344 turbine system without any controller implementation, which include the variation history comparisons of the
345 PMSG speeds and the wind turbine main shaft torques. To a given electric resistance load, the PMSG
346 reference speed is set to 190 rad/s. From Fig. 11, it is clearly seen that, during low wind speed periods, the
347 PMSG can obtain additional driving torque from small-scale CAES integration, thus the simulated hybrid
348 system with the PI controller connected can compensate the required electric power efficiently; meanwhile,
349 during high wind speed periods, the designed system with the fuzzy logic controller activated can track the
350 reference speed very well, which may reveal only few extra load and tiny inertia added by the mechanical
351 power transmission. Furthermore, it looks that the performance of PI control to the scroll expander supply
352 pressure is not as good as that of the pitch angle fuzzy logic control. This could be resulted from the high
353 nonlinearity characteristics of the compressed air.

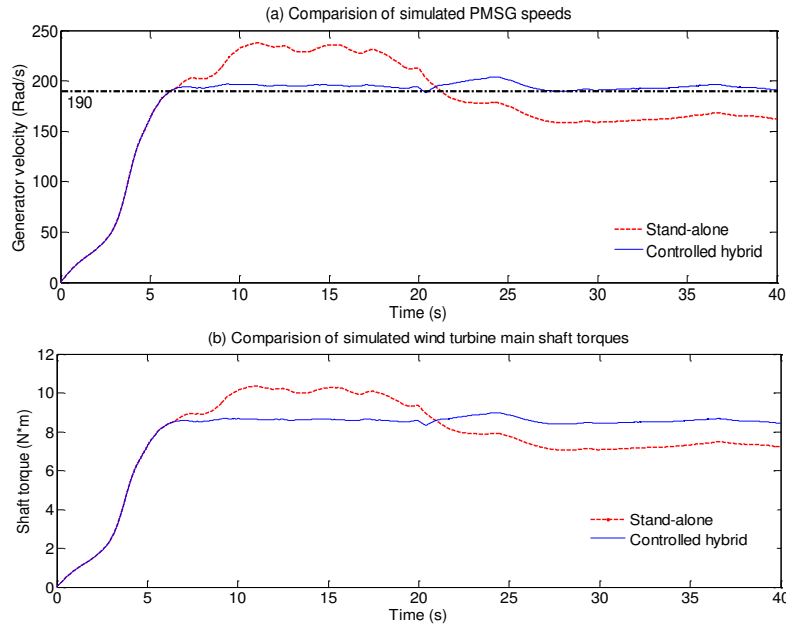


Fig. 11 Comparison of dynamic responses of the hybrid system with designed controllers connected and the stand-alone wind turbine without any control

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Correspondingly, Fig. 12 shows the simulation results of the dynamic responses of the controlled variables in the hybrid system when the input wind speed profile is given as Fig. 10. In the designed system, the pitch angle and the supply pressure of the scroll expander are the controlled variables. From Fig. 12 (a), it can be observed that, under the conditions of high wind speed, the pitch angle varies within the range from 0 to 11 degrees. This is because the pitch angle is adjusted by fuzzy logic control to maintain a relatively steady wind power output for the PMSG speed getting close to the required reference (190 rad/s). At low speed situations, the pitch angle is set to 0 degree for maximizing the capability of wind turbine blades to extract the wind energy. In Fig. 12 (b), the variation history of the supply pressure of scroll expander is presented. The expander is activated at 20 seconds from the time at which the wind turbine are in operation. Once Clutch B is engaged, its supply pressure is always managed by the designed PI controller. From the simulation study, the scroll expander can speed up very quickly after its activation, mainly due to the expander small inertia.

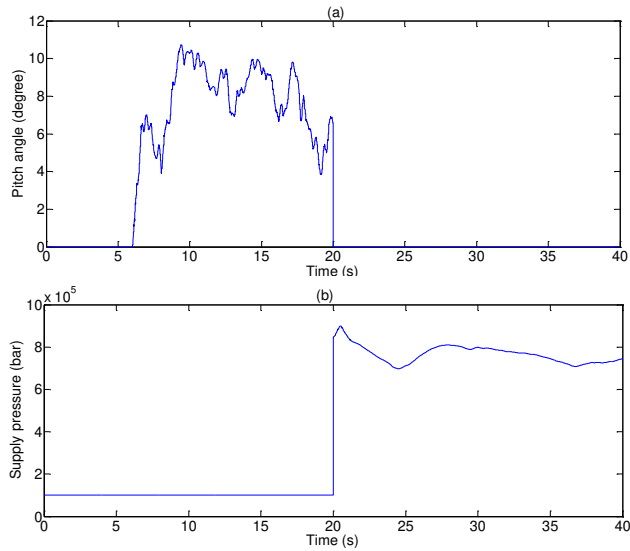


Fig. 12 Dynamic responses of the controlled variables in the hybrid system

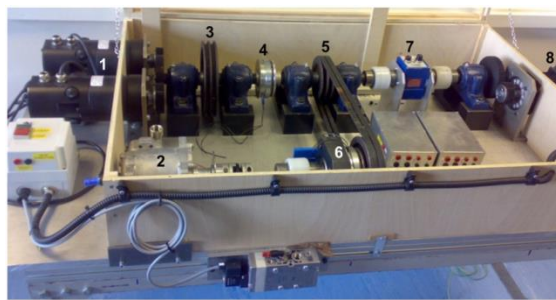
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373 5. Experimental Tests

374 An experimental test rig corresponding to the designed hybrid system is built in the authors' research
 375 laboratory at the University of Warwick, as shown in Fig. 13. The block diagram of this prototype test rig is
 376 illustrated in Fig. 14. Due to the limitation of indoor laboratory work, the test rig uses a "Wind Turbine
 377 Simulator (WTS)" to replace the practical wind turbine blade, which consists of dual DC motors, their power
 378 supplies and some auxiliaries (Fig. 14). The function of this simulator is to mimic the real fluctuant turbine
 379 blade torque and wind power scenario. The reasonability of WTS had been proven in some literatures, e.g.,
 380 [32, 33].

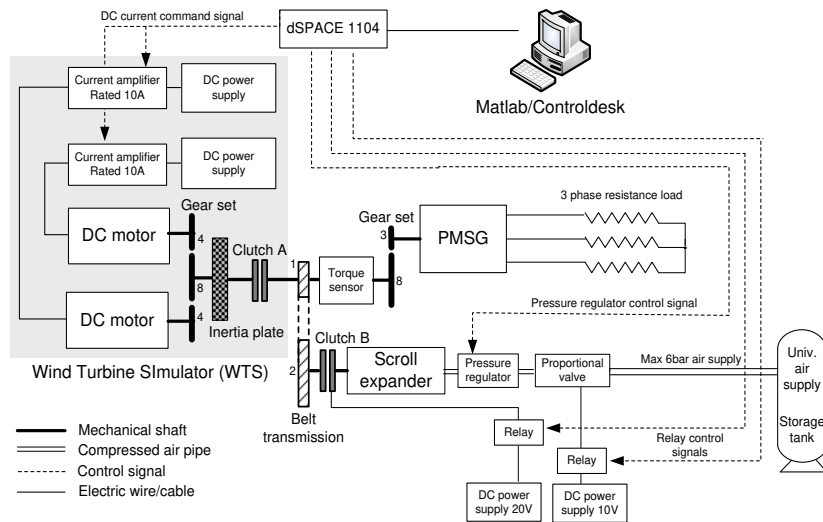


1 – DC motor (wind turbine simulator) 5 – Belt transmission system
 2 – Scroll expander 6 – Clutch B
 3 – Inertia plate 7 – Torque and speed meter
 4 – Clutch A 8 – PMSG

381

382

Fig. 13 The experimental test rig of the wind turbine system integrated with small-scale CAES



383

384

Fig. 14 Block diagram of the test rig of the wind turbine system integrated with small-scale CAES

385

Based on Fig. 14, the main test system components are listed in Table 3. The test rig consists of dual DC
 386 motors, a PMSG and its 3-phase resistance load, a scroll expander, a compressed air storage tank, a belt
 387 transmission subsystem, two clutch mechanisms, two gear sets, a pneumatic valve, a digital pressure regulator,
 388 DC power supplies, sensors and meters for electrics and pneumatics, etc. The employed scroll expander is
 389 modified from a scroll compressor (Table 3). In addition, two gears sets (Fig. 14) as speed match devices are
 390 applied to ensure that each facility can work around at its rated condition. The speed ratios to the gear sets are
 391 determined by the rated speeds of these facilities.

392

Table 3 Machines for the experimental test system

Name	Serial number/description	Manufacturer
DC motor	SN:M4-2952X-2100t-000	Callan Tech.
PMSG	SN:SGMSS-20A	Yaskawa Elec.
Scroll expander	Modified from compressor TRSA090	Sanden
Air tank	Max 6 bar from univ. storage tank	BOC UK
Controller	Model: RTI1104	dSPACE
Clutch A	SN:CS-10-31G, 24V	Mikipulley
Clutch B	101-10-15G, 24V	Mikipulley
DC power supply	90Vdc, 0-10A dc, for DC motor	TRM Elec.
DC amplifier	SN:10/100, 24-100Vdc, 0-10A dc	TRM Elec.
Voltage transducer	SN:LV 25-P, ±10V	LEM
Current transducer	SN:LTSR 15-NP, ±15A	LEM
Pressure sensor	SN:SDE1-D10-G2-W18-L-PU-M8	FESTO
Flow meter	SN:MS6 SFE-F5-P2U-M12	FESTO
Pressure regulator	SN:VPPM-6L-L-1-G18-0L10H-V1N	FESTO
Pneumatic valve	SN:MYPE-5-1/4-010-B, 0-10 bar	FESTO
Torque sensor	SN:RWT 310, 0-2000 RPM	Sensor Tech.
Temperature sensor	K-type thermocouple	RS UK

393

394 A dSPACE real-time controller (Model: RTI1104) is chosen for collecting the experimental data from
 395 electric and pneumatic sensors. The experimental data is monitored and collected in dSPACE
 396 Controldesk/Matlab environment. This real-time controller is also used for controlling the whole hybrid
 397 system operations, which include activating the scroll expander, engaging Clutch B (Clutch A is always
 398 engaged as described in Section 2), implementing PI control to the pressure regulator and managing the WTS
 399 torque output to the hybrid system (refer to Fig. 14).

400 Fig. 15 illustrates the block diagram of the WTS in the hybrid system test rig. The WTS can be considered
 401 as a variant of Rapid Control Prototyping (RCP) – using simple PID control to command dual DC motors for
 402 mimicking the real wind turbine behaviours. Similarly to the simulation study, a white noise source with a
 403 shaping filter is used to generate the wind speed profile ([32-34]). A typical torque-current closed loop PID
 404 control is implemented on the DC motors as shown in Fig. 15.

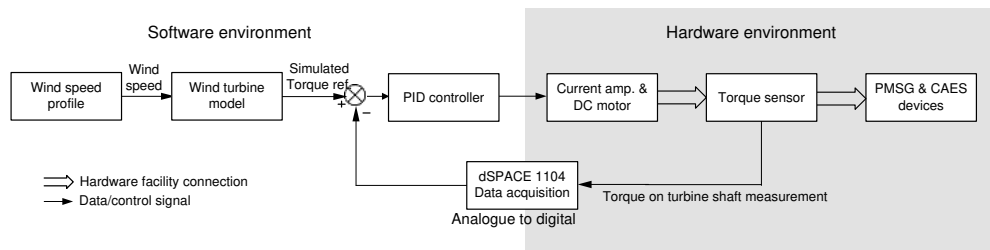


Fig. 15 Block diagram of the WTS in the hybrid system test rig

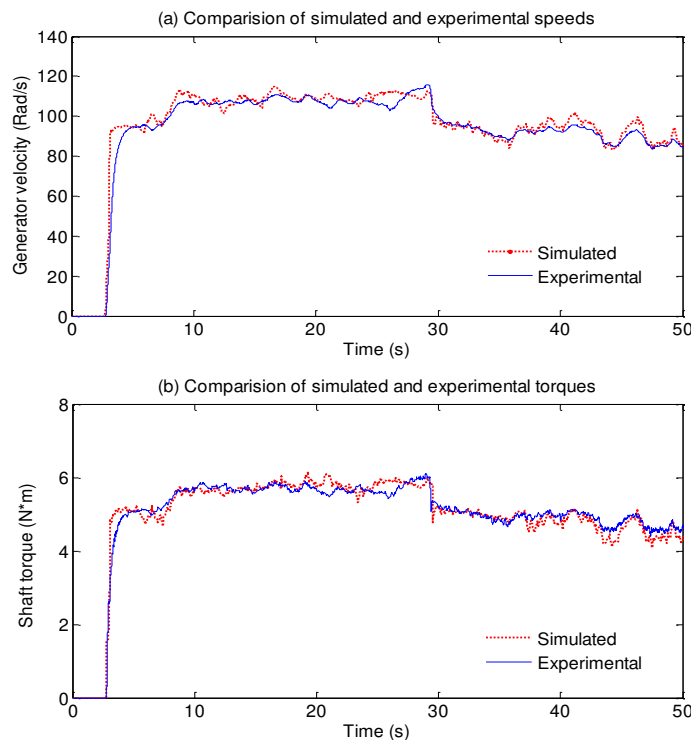


Fig. 16 Comparisons of the test results and the simulated data for the WTS

410 The comparisons of the experimental test results and the simulated data for the WTS are given in Fig. 16,
 411 including the wind turbine shaft speeds and torques respectively. It can be seen that the experimental dynamic
 412 performance of the WTS torque can track well to that of the simulated torque reference. From the experimental
 413 tests, it is proven that the WTS can meet the laboratorial requirements to mimic the practical scenarios of wind
 414 power generation systems.

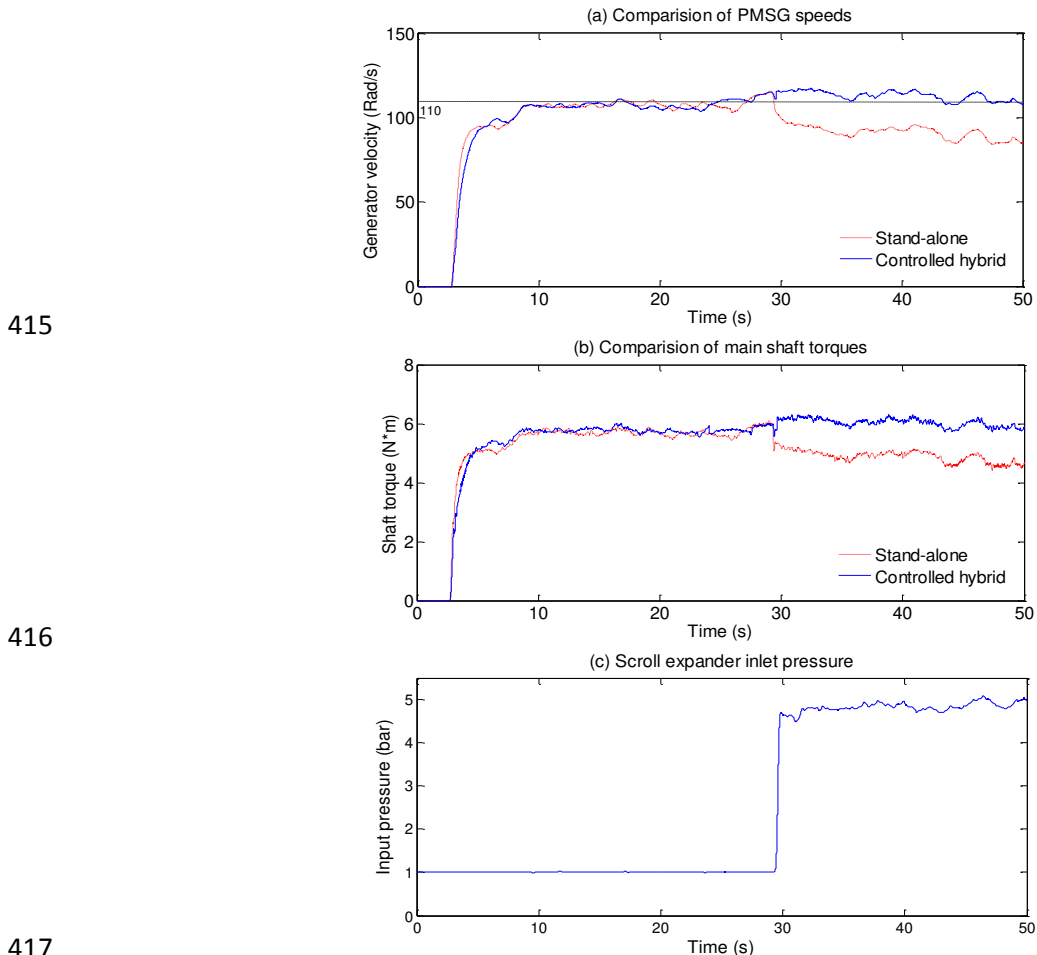


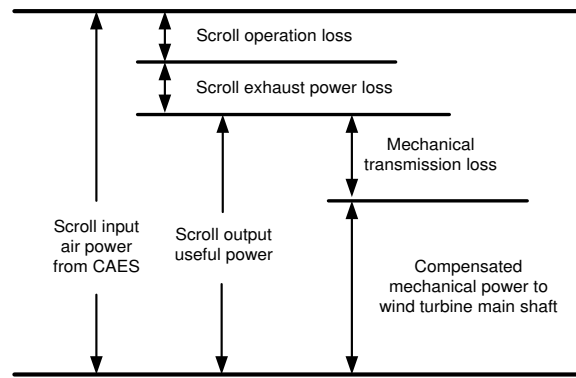
Fig. 17 Experimental test results of the stand-alone wind turbine and the controlled hybrid system

419 Due to the laboratory limitations and the university safety regulations, the experimental test to the hybrid
 420 system with the control strategy implemented mainly focuses on the study of system operation under low wind
 421 speed situations. It is for observing the test rig dynamic responses at the moment of Clutch B engagement and
 422 the controller performance to the CAES subsystem for compensation work. Fig. 17 shows the comparisons of
 423 the experimental test results of the hybrid system under the wind turbine stand-alone mode and the wind
 424 turbine integrated CAES with controller connected mode. The speed reference is set to 110 rad/s. It can be
 425 seen that, with the CAES integration and the scroll expander supply pressure control, the designed
 426 experimental test system can maintain relatively steady outputs and to meet the speed reference under the low

427 wind speed conditions. The CAES system can contribute controllable mechanical power to the hybrid wind
 428 turbine system for generating required electricity. Thus the experiment results shown in Fig. 17 verify that the
 429 idea proposed in this paper is feasible and the corresponding prototype can work properly.

430 6. Efficiency analysis

431 To the situation of controlled hybrid system operation, the power transmission and conversion from a small-
 432 scale CAES to the wind turbine system is schematically illustrated in Fig 18. The scroll expander converts the
 433 energy extracted from the stored compressed air into the useful mechanical energy which is in turn transferred
 434 to the wind turbine main shaft through the belt transmission system. For such process, energy losses are
 435 inevitable, such as the scroll expander operation loss due to friction, vibration, air leakage, lubricant viscosity,
 436 etc.



437
 438 Fig.18 Power transmission and conversion from the small-scale CAES to the wind turbine system

439 The power efficiency of the engaged CAES system in this paper is defined as,

$$440 \quad \eta_{eff} = \frac{\text{Increased mechanical power resulted from CAES compensation}}{\text{Input compressed air power from CAES}} \quad (36)$$

441 This power efficiency reveals the performances of the designed small-scale CAES facility and the mechanical
 442 power transmission. In addition, the increased mechanical power resulted from CAES compensation is
 443 considered as the difference of the hybrid system main shaft powers between the stand-alone mode and hybrid
 444 mode under the same driving conditions.

445 From the above description, it is necessary to quantitatively analysis how much air power/energy carried by
 446 compressed air enters into the scroll expander. One simplified approach is adopted for calculating the input air
 447 power referred to STP (Standard Temperature and Pressure, 0 °C at 1 bar), which is ([35, 36]):

448
$$\dot{Q}_{in} = \dot{m}_{in} RT_{atm} \left[\ln \frac{P_{in}}{P_{atm}} + \frac{k}{k-1} \left(\frac{T_{in}}{T_{atm}} - 1 - \ln \frac{T_{in}}{T_{atm}} \right) \right]$$
 (37)

449 where \dot{Q}_{in} is the input air power to the scroll expander, \dot{m}_{in} is the input air mass flow rate, T is temperature, p
 450 is pressure, k is the specific heat ratio, subscript *atm* means atmospheric state and *in* is inlet thermodynamic
 451 state. When the environment shifts 100 K from the atmospheric temperature, the temperature variation to the
 452 change of air power is limited ([35, 36]). Thus it can assume $T_{in} = T_{atm}$ and then substituting this into Eq. (37),
 453 the air power can be calculated by,

454
$$\dot{Q}_{in} = \dot{m}_{in} RT_{atm} \ln \frac{P_{in}}{P_{atm}} = P_{atm} w_{in} \ln \frac{P_{in}}{P_{atm}}$$
 (38)

455 where w_{in} is the input volumetric air flow rate.

456 Building upon the available sensors in the laboratory for data acquisition, three groups of experimental tests
 457 of the hybrid system are implemented to analyse the power efficiency. The experimental results are given in
 458 Table 4. The tests are conducted under the condition of maintaining the Wind Turbine Simulator (WTS) power
 459 output at three different levels, which result in three measured power levels of shaft power under the wind
 460 turbine stand-alone mode, which are 640, 510 and 390 watts respectively (Table 4). Also, in each group of the
 461 tests, the hybrid system is operated at the different inlet air pressures to the scroll expander for the power
 462 efficiency comparison.

463 Table 4 Power efficiency analysis based on the experimental data

Inlet pressure (bar)	Inlet flow rate (L/min)	Scroll inlet temperature (°C)	Scroll outlet temperature (°C)	Stand-alone power (W)	Hybrid power (W)	Power Efficiency (%)
5.76	335	24.5	17	640	980	34.76%
4.90	255	24.5	17.5	640	950	45.87%
3.95	230	24.5	18	640	900	49.34%
5.75	340	24.5	17.2	510	925	41.84%
4.88	250	24.5	17.6	510	875	55.27%
3.90	220	24.5	18.1	510	780	54.07%
5.73	320	24.5	17.2	390	800	44.01%
4.90	245	24.5	17.7	390	690	46.20%
3.90	205	24.5	18.1	390	610	47.28%

464
 465 From the experimental results, it can be found that, in each group of the tests, the variations of the power
 466 efficiency are relevant to the scroll expander inlet air pressure, which indicates that the air pressure and mass

467 flow rate should be well managed and controlled to achieve higher efficiency. According to this, under a given
468 working condition, it may suggest using lower inlet pressures of compressed air for obtaining higher power
469 efficiencies. From Table 4, the maximum power efficiency is around 55%. The main reason for this moderate
470 efficiency is that CAES and pneumatic drives have relatively low efficiencies in general. From the reported
471 figures, in most cases, around 20-30% energy efficiency can be achieved for pneumatic actuator (drive)
472 systems; 45-54% cycle efficiency has been reported for the existing commercialized large-scale CAES plants
473 [9, 22, 35, 36]. Although the scroll expander has relatively higher energy conversion ability compared to
474 traditional pneumatic actuators ([23, 24]), the efficiency related to CAES and its components is still a key issue
475 which needs further research and development. Also, considering the moderate efficiency of CAES and
476 pneumatic drives, the power efficiency analysis indicates that the designed mechanical power transmission for
477 the hybrid system can deliver an acceptable performance.

478 **7. Conclusion**

479 In this paper, a new concept of hybrid system is proposed, which consists of a kW-level wind turbine
480 integrated with a small-scale CAES unit. To avoid mechanical force coupling between the driving torques
481 from wind power and the air expander, a mechanical transmission mechanism is developed to smoothly
482 integrate the two torque sources. The complete dynamic mathematical model of the hybrid system is
483 developed and implemented in Matlab/Simulink simulation environment for comprehensive simulation study
484 of system dynamic behaviours. An appropriate control strategy is developed for the system to smooth the
485 transient fluctuations and compensate the energy gap of wind power generation. A prototype system is built
486 and installed in the research lab for verifying the design idea. From both the simulation and test results, the
487 hybrid wind turbine system can generate the reliable steady power output with the compensation power from
488 CAES. It can be concluded that the proposed hybrid system of wind turbines and CAES is feasible with a great
489 potential for future industrial applications.

490 The energy conversion efficiency from the compressed air energy to the electrical power output has been
491 investigated with various operation conditions. The system test results indicated that the efficiency can be up
492 to 55% under a well-controlled operation condition, which is higher than the typical pneumatic actuator
493 efficiency. The findings have provided essential evidences and information for the next stage of research
494 which will lead to improved efficiency and reliability.

495

496 **Acknowledgement:**

497 The authors would like to thank Advantage West Midlands and the European Regional Development Fund,
498 funders of the Science City Research Alliance Energy Efficiency Project – a collaboration between the
499 Universities of Birmingham and Warwick and UK Engineering and Physical Sciences Research Council
500 (EPSRC, EP/K002228/1) for the funding support.

501

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