Contents lists available at ScienceDirect

Applied Energy

journal homepage: www.elsevier.com/locate/apenergy

A near-isothermal expander for isothermal compressed air energy storage system

Xinjing Zhang^{a,b}, Yujie Xu^{a,b}, Xuezhi Zhou^a, Yi Zhang^{a,b}, Wen Li^{a,b}, Zhitao Zuo^{a,b}, Huan Guo^a, Ye Huang^c, Haisheng Chen^{a,b,*}

^a Institute of Engineering Thermophysics, Chinese Academy of Sciences, 100190 Beijing, China

^b University of Chinese Academy of Sciences, 100190 Beijing, China

^c School of the Built Environment, University of Ulster, Newtownabbey BT37 OQB, UK

HIGHLIGHTS

• A specific expander was analysed by theoretical and experimental methods.

• A quasi-isothermal expansion process was further improved by spraying water into the cylinder.

• Specific work generation was enhanced and the cylinder size was reduced.

ARTICLE INFO

Keywords: Compressed air energy storage Isothermal expander Experiment and simulation Specific work High pressure ratio

ABSTRACT

Compressed air energy storage technology is considered as a promising method to improve the reliability and efficiency of the electricity transmission and distribution, especially with high penetration of renewable energy. Being a vital component, the expander takes an important role in compressed air energy storage operation. The specific work of an expander can be improved through an isothermal expansion compared with the adiabatic expansion process due to a nearly constant temperature which enables the expander to operate with a high pressure ratio. In this study, a specific reciprocating expander with a high pressure ratio was developed and its adiabatic expansion characteristics were measured. Numerical modelling was performed to simulate adiabatic expansion. This model was also validated by experimental results. Based on these findings, we propose a quasi-isothermal expansion process by introducing water-air direct heat transfer equations. Simulation results showed that when spraying tiny water droplets into the cylinder, the specific work generated was improved by 15.7% compared with that of the adiabatic expansion under the same air mass flowrate, whilst the temperature difference was only about 10% of that of the adiabatic process, and cylinder height was decreased by 8.7%. The influence of water/air mass flowrate ratio and the inlet temperature on the expander performance was also studied.

1. Introduction

As clean energy is vital to the mission of mitigating climate change and air pollution, the business case for clean energy is growing, and the trend toward a cleaner power sector is supported by beneficial relationships between economies and emissions, born out in relationship statistics (economies grow, emissions fall), private-sector emissions reductions, and market forces in the power sector and global momentum on emission-reducing technologies [1]. Renewable energy sources (e.g. solar and wind) are vitally important alternatives for clean, affordable, and reliable energy paradigms. The cost of electricity generated from renewables fell dramatically between 2008 and 2015: down 41% for wind, 54% for rooftop solar photovoltaic (PV) installations, and 64% for utility-scale PV [1]. Global capital investment in these clean energy resources was twice as much as that of fossil fuels in 2015 [1,2].

The mismatch between an intermittent electricity supply and demand over multiple time and energy scales necessitates energy storage to balance and optimize power flow and generation. Electrical energy storage can play an important role in decarbonizing the electricity

https://doi.org/10.1016/j.apenergy.2018.04.055 Received 30 December 2017; Received in revised form 4 April 2018; Accepted 26 April 2018





^{*} Corresponding author at: Institute of Engineering Thermophysics, Chinese Academy of Sciences, 100190 Beijing, China. *E-mail address*: chen_hs@iet.cn (H. Chen).

^{0306-2619/ © 2018} Elsevier Ltd. All rights reserved.

SymbolsConcepts, Units W_{iso} Ideal work generation of isothermal expansion, kJAopen area of the inlet/outlet valve, m2 W_{pump} Work consumption of water pump, kJBDCbottom dead center $Greek symbols$ c_p specific heat at constant pressure, kJ/(kgk) α heat transfer coefficient c_V specific heat at constant volume, kJ/(kgk) α heat transfer coefficient $CAES$ compressed air energy storage η_{adia} Isentropic efficiency d water droplet diameter, m η_{isoth} isothermal efficiency considering pump power consump- tion g acceleration, m/s2 κ specific heat ratio h craptific enthow, k l/(rg κ specific heat ratio	Nomenclature		W_{n_iso}	Work generation of near-isothermal expansion, kJ	
SymbolsConcepts, Units W_{pump} Work consumption of water pump, kJAopen area of the inlet/outlet valve, m2Work consumption of water pump, kJBDCbottom dead centerGreek symbols c_p specific heat at constant pressure, kJ/(kgk)a c_v specific heat at constant volume, kJ/(kgk)aCAEScompressed air energy storage η_{adia} G_D drag coefficient η_{isoth} dwater droplet diameter, m η_{isoth} Dcylinder diameter, m η_{isoth} gacceleration, m/s2 κ specific heat ratio λ kspecific heat ratio			Wiso	Ideal work generation of isothermal expansion, kJ	
Aopen area of the inlet/outlet valve, m^2 BDCbottom dead centerGreek symbols c_p specific heat at constant pressure, kJ/(kgk)aheat transfer coefficient c_v specific heat at constant volume, kJ/(kgk) α heat transfer coefficientCAEScompressed air energy storage η_{adia} Isentropic efficiency G_D drag coefficient η_{isoth} isothermal efficiencydwater droplet diameter, m η_{isoth} isothermal efficiency considering pump power consump-Dcylinder diameter, m η_{isoth} tiongacceleration, m/s^2 κ specific heat ratiohcranging on the hyse kL/krg λ ratio of connected red length to crank radius	Symbols	Concepts, Units	W_{pump}	Work consumption of water pump, kJ	
BDCbottom dead centerGreek symbols c_p specific heat at constant pressure, kJ/(kgk) α heat transfer coefficient c_v specific heat at constant volume, kJ/(kgk) α heat transfer coefficientCAEScompressed air energy storage η_{adia} Isentropic efficiency C_D drag coefficient η_{isoth} isothermal efficiency d water droplet diameter, m η_{isoth} isothermal efficiency considering pump power consump- D cylinder diameter, mtion g acceleration, m/s ² κ specific heat ratio h crantific ontholy $k L/kg$ λ	Α	open area of the inlet/outlet valve, m^2			
c_p specific heat at constant pressure, kJ/(kgk) α heat transfer coefficient c_v specific heat at constant volume, kJ/(kgk) α heat transfer coefficientCAEScompressed air energy storage η_{adia} Isentropic efficiency C_D drag coefficient η_{isoth} isothermal efficiency d water droplet diameter, m η_{isoth} isothermal efficiency considering pump power consump- D cylinder diameter, mtion g acceleration, m/s ² κ specific heat ratio h creatific ontholy, kL/kg λ	BDC	bottom dead center	Greek sy	Greek symbols	
c_v specific heat at constant volume, kJ/(kgk) α heat transfer coefficientCAEScompressed air energy storage η_{adia} Isentropic efficiency C_D drag coefficient η_{isoth} isothermal efficiency d water droplet diameter, m η_{isoth} isothermal efficiency considering pump power consump- D cylinder diameter, mtion g acceleration, m/s ² κ specific heat ratio h creation of connected rod length to creatk radius	c_p	specific heat at constant pressure, kJ/(kgk)			
CAEScompressed air energy storage η_{adia} Isentropic efficiency C_D drag coefficient η_{isoth} isothermal efficiency d water droplet diameter, m η_{isoth2} isothermal efficiency considering pump power consump- tion g acceleration, m/s ² κ specific heat ratio h crastific enthely, k L/kg λ ratio of connected rod length to crask radius	c_{v}	specific heat at constant volume, kJ/(kgk)	α	heat transfer coefficient	
C_D drag coefficient η_{isoth} isothermal efficiencydwater droplet diameter, m η_{isoth} isothermal efficiency considering pump power consump-Dcylinder diameter, mtiongacceleration, m/s ² κ specific heat ratiohcrastific onthelay, kL/kg λ ratio of connected rod length to crank radius	CAES	compressed air energy storage	η_{adia}	Isentropic efficiency	
dwater droplet diameter, m η_{isoth2} isothermal efficiency considering pump power consump-Dcylinder diameter, miongacceleration, m/s ² κ specific heat ratiohcrossife onthology, kL/kg λ ratio of connected rod length to crank radius	C_D	drag coefficient	η_{isoth}	isothermal efficiency	
Dcylinder diameter, mtiongacceleration, m/s^2 κ specific enthology, kL/kg λ ratio of connected rod length to crank radius	d	water droplet diameter, m	η_{isoth2}	isothermal efficiency considering pump power consump-	
g acceleration, m/s ² κ specific heat ratio h specific onthology kL/kg λ ratio of connected rod length to graph radius	D	cylinder diameter, m		tion	
h specific onthology kL/kg and the specific onthology of connected rod length to crank radius	g	acceleration, m/s ²	к	specific heat ratio	
λ_e fails of connected for length to train factors	h	specific enthalpy, kJ/kg	λ_e	ratio of connected rod length to crank radius	
<i>ICAES</i> isothermal compressed air energy storage λ thermal conductivity, W/(m·K)	ICAES	isothermal compressed air energy storage	λ	thermal conductivity, W/(m·K)	
m_a mass of air, kg/s μ gas flow coefficient	m_a	mass of air, kg/s	μ	gas flow coefficient	
m_w mass of water, kg/s μ_a dynamic viscosity of the air, Pas	m_w	mass of water, kg/s	μ_a	dynamic viscosity of the air, Pa·s	
<i>Nu</i> Nusselt number ρ density, kg/m ³	Nu	Nusselt number	ρ	density, kg/m ³	
<i>P</i> pressure, Pa φ crank angle, rad	Р	pressure, Pa	φ	crank angle, rad	
Pr Prandtl number ω angular speed, rad/s	Pr	Prandtl number	ω	angular speed, rad/s	
Q heat exchange through the cylinder walls, kJ ψ flow function	Q	heat exchange through the cylinder walls, kJ	Ψ	flow function	
Q_c transferred heat between water and air, kJ	Q_c	transferred heat between water and air, kJ			
S stroke distance, m Subscripts	S	stroke distance, m	Subscrip	ts	
S_w water droplets surface area, m ²	S_w	water droplets surface area, m ²			
T temperature, K a air	Т	temperature, K	а	air	
<i>TDC</i> top dead center <i>A</i> outlet for air phase, accumulated water on the piston for	TDC	top dead center	Α	outlet for air phase, accumulated water on the piston for	
t time, s water phase	t	time, s		water phase	
<i>u</i> specific internal energy, kJ/kg <i>E</i> inlet	и	specific internal energy, kJ/kg	Ε	inlet	
V volume, m ³ I before the inlet/outlet value	V	volume, m ³	Ι	before the inlet/outlet valve	
<i>v</i> velocity, m/s II after the inlet/outlet valve	ν	velocity, m/s	Π	after the inlet/outlet valve	
w velocity, m/s <i>out</i> outlet	w	velocity, m/s	out	outlet	
W mechanical energy, kJ pump pump	W	mechanical energy, kJ	pump	pump	
<i>W_{adia}</i> Work generation of adiabatic expansion, kJ <i>w</i> water	Wadia	Work generation of adiabatic expansion, kJ	w	water	
<i>W</i> _{s_adia} Ideal work generation of adiabatic expansion, kJ	W_{s_adia}	Ideal work generation of adiabatic expansion, kJ			

sector by offering a new, carbon-free source of operational flexibility, improving the utilization of generation assets and facilitating the integration of variable renewable energy sources [2,3]. Low-cost fabricated compressed air energy storage (CAES) will be a most promising method to store electricity for medium- and long-term periods [2]. When off-peak electricity is available it can be used to produce compressed air via a series of compressors. Compressed air is then stored in a reservoir. During peak periods the stored compressed air is released to drive expanders to generate electricity [4,5]. CAES technologies can help accommodate fluctuations in wind generation and decrease transmission line size rather than enlarging lines to match maximum power levels. The first patent for CAES technology was filed by Frazer W. Gay in 1948 [6]. This technology has been developed since the 1970s as a load-following and load-peaking power system. CAES system has an estimated efficiency of 70% with an expected lifetime of about 40 years [7]. Two commercial CAES plants have been constructed in Huntorf, Germany and McIntosh, USA [8]. Other countries such as the UK, Denmark, and the Netherlands have been keen to develop CAES plants as well [4].

Many types of CAES have been studied and developed, including conventional CAES, advanced-adiabatic CAES, liquid air energy storage, isothermal CAES, and so forth [4]. In the ideal isothermal CAES (ICAES) process, the temperature during compression is kept constant while related heat is released. The power required to run the compressor is correspondingly lower than that required to run an adiabatic compressor with the same pressure ratio. During expansion, related heat is supplied continuously to ensure expansion at a constant temperature. Thus, the electrical power used to run the compressor during charging can be completely recovered during discharging. The ideal cycle efficiency of ICAES systems can be as high as 100% [4,9]. In this study, a liquid piston based ICAES is proposed, which can yield a compressive efficiency of 89.0% according to the simulation results [8].

The compressor and expander are the pivotal components of the CAES system [10]. Generally, most gas power cycles perform work by expanding adiabatically and produce less specific work than their isothermal counterparts, which run at constant temperature. Isothermal expansion leads to a high pressure ratio and high power density, improving specific work generation. Meanwhile, lower inlet/outlet air temperature differences result in better performance in low-grade heat applications [11]. If heat is continuously transferred to the working fluid during the expansion process, this results in isothermal expansion or, more accurately, quasi-isothermal expansion, which may be somewhere between the adiabatic and ideal isothermal processes [12]. Compressing and expanding a gas nearly isothermally allows efficiency losses due to temperature deviations to be minimized or eliminated, which can in turn prevent heat transfer loss leading to improved efficiency [13]. An experimental system using condensable gas of R134a was built to investigate energy storage potential and compression/expansion characteristics. The experimental results showed a round trip efficiency of 95.8% was achievable [13]. Cicconardi et al. proposed using many super-heaters to make the expansion gradually approach isothermal conditions in a steam power plant and found that the thermal efficiency can be improved from 38.5% to 49.2% [14]. Kim et al. indicated that multi-stages compression with intercooling and multi-stages expansion with reheating could also transform the adiabatic process into a near-isothermal process. Thus, the exergy losses due to heat transfer were decreased by minimizing the temperature differences between heat exchangers. System efficiency was as high as 71.6% [15]. Woodland et al. [16] carried out theoretical study of the Organic Rankine Cycle (ORC) based on a liquid flooding expander. For

Download English Version:

https://daneshyari.com/en/article/6680041

Download Persian Version:

https://daneshyari.com/article/6680041

Daneshyari.com