ELSEVIER
Available online at www.sciencedirect.com
SciVerse ScienceDirect
Energy Procedia 00 (2016) 000–000
Energy Procedia
www.elsevier.com/locate/procedia

The 8th International Conference on Applied Energy – ICAE2016

Modelling of plate heat exchangers and their associated CO_2 trancritical power generation system

Y.T. Ge[a]*, L. Li[a], X. Luo[b], S.A. Tassou[a]

[a]*RCUK National Centre for sustainable Energy Use in Food Chains (CSEF), Institute of Energy Future,Brunel University London, Uxbridge, Middlesex, UB8 3PH, UK*
[b]*National Key Laboratory of Science and Technology on Aero Engines Aero-thermodynamics, The Collaborative Innovation Centre for Advanced Aero-Engine of China, Beihang University, Beijing 10191, China.*

Abstract

Globally, there is no shortage of low-grade waste and renewable heat sources that can be converted into electricity and useful heat using applicable thermodynamic power cycles and appropriate working fluids. As a natural working fluid, CO_2 is a promising candidate for application in low-grade power generation systems but require optimised design and evaluation.

Since CO_2 working fluid has a low critical temperature (31.1°C) and high critical pressure (73.8 bar), a CO_2 low-grade power generation system will most likely undergo supercritical Rankine (T-CO_2) cycles. A T-CO_2 system normally consists of a CO_2 supercritical gas heater, expander, recuperator, condenser and liquid pump with the CO_2 gas heater being a crucial component in determining system thermal and exergy efficiencies. In this paper, the models of a thermal oil-CO_2 plate gas heater has been developed and validated with measurements of a 5kW$_e$ T-CO_2 system test rig. The model is then integrated with other system component models to establish the system model. The system model can be used to evaluate and compare system performances at different operating conditions, including variable CO_2 gas heater pressures and heat sink parameters, thereby optimising system operations.

Keywords: Low-grade power generation ; CO_2 Transcritical Rankine Cycles; CO_2 gas heater model and validation; Simulation

* Corresponding author. Tel.: +44-1895-266722; fax: +44-1895-256392.
E-mail address: yunting.ge@brunel.ac.uk.

1. Introduction

Over 50% of total applicable industrial waste heat and renewable energy can be classified as low-grade heat sources with a temperature below 230°C [1]. In order to conserve valuable fossil fuels, it is therefore imperative to convert this waste heat into electricity for power generation. Currently, an Organic Rankine Cycle (ORC) is a practicable thermodynamic process for converting low-grade heat into electricity. However, most of the ORC working fluids are HFCs such as R245fa, whose features include zero Ozone Depletion Potential (ODP) but comparatively high Global Warming Potential (GWP). In addition, a HFC ORC has a constant evaporating temperature for its high pressure heat addition process. This will cause a temperature profile mismatch between the ORC working fluid and sensible waste heat flow, thus increasing irreversible losses and diminishing system efficiency. Therefore, a low-grade power generation system with applicable thermodynamic power cycle and appropriate working fluid needs to be further investigated.

Nomenclature

A	heat transfer area (m^2)	Subscripts	
C	product of mass flow rate and specific heat (W/K)	air	air
CP	constant pressure specific heat of air (J/kg.K)	aircd	condenser air inlet
L	vertical length of heat exchanger (m)	ci	cold fluid inlet
$\dot{m}$	mass flow rate (kg/s)	co	cold fluid outlet
N	number of plate	exp	expander
P	pressure (pa)	expin	expander inlet
Q_T	heat capacity (W)	hi	hot fluid inlet
T	temperature (^{0}C)	ho	hot fluid outlet
ΔT	temperature difference (K)	max	maximum
U	heat transfer coefficient ($W/m^2.K$)	min	minimum
UA	overall heat conductance (W/K)	oil	thermal oil
W	horizontal length of heat exchanger (m)	p	plate
ε	heat transfer effectiveness (-)	sc	subcooling
ϕ	length ratio		
α	heat transfer coefficient ($W/m^2.K$)		
β	chevron angle		
δ	plate thickness (mm)		

Alternatively, carbon dioxide is an environmentally friendly working fluid with zero ODP and negligible GWP (<1.0). It also has superb thermophysical properties and features of being non-flammable, non-toxic and thermally stable. On the other hand, should CO_2 be applied into a low-grade power system, the thermodynamic power cycle is highly likely in supercritical Rankine (T-CO_2) cycles considering its high critical pressure (73.9 bar) but low critical temperature (31.1°C). Research results showed that a T-CO_2 cycle was more suitable to low-grade power generation in term of thermal efficiency [2]. The supercritical heat addition process in a T-CO_2 can produce matched temperature profiles between the working fluid and sensible heat source and thus reduce the irreversibility involved. As such, a T-CO_2 low-grade power generation system is expected to be more thermally efficient than that of a conventional ORC [3].

A T-CO_2 power generation system consists of a number of essential components including a gas heater, expander, recuperator, condenser and liquid pump. Due to the high critical pressure of CO_2, the pressure of heating processes in the T-CO_2 Rankine cycles would also be high (typically above 90 bar) such that special designs are greatly required for system components such as heat exchangers, liquid pumps and expanders. Consequently, up till now, investigations on low temperature heat source energy conversion systems with T-CO_2 Rankine cycles have been limited to small-scale laboratory work and theoretical analyses. For instance, more effort has been placed upon modelling analyses in various T-CO_2 Rankine cycles with the application of solar-CO_2 power generation [4] and low temperature waste heat recovery [5]. However, to fully understand the effect of heat source parameters on the T-CO_2 system performance, comprehensive heat transfer calculation and analysis of a CO_2 gas heater need to be further investigated.

In this paper, a detailed model of a thermal oil heated CO_2 plate gas heater is developed and validated with measurements. The model is then integrated with other component models in a T-CO_2 system so as to predict the effects of the gas heater pressures and heat sink parameters on system performance. These are helpful to understand the system operation and thus optimise designs and controls.

2. Model development and validation

A test rig of the T-CO_2 power generation system was set up at Brunel University, as schematically shown in Fig 1. To facilitate analysis, the corresponding T-S diagram for the T-CO_2 Rankine cycle (with recuperator) is also depicted in Fig 1. The test rig consisted of various components including a CO_2 turbine/expander, electricity generator, recuperator, air-cooled condenser, receiver, liquid pump and thermal oil heated CO_2 gas heater. Some initial measurements were carried out with varied heat source and sink parameters which are used to validate the models developed in this paper.

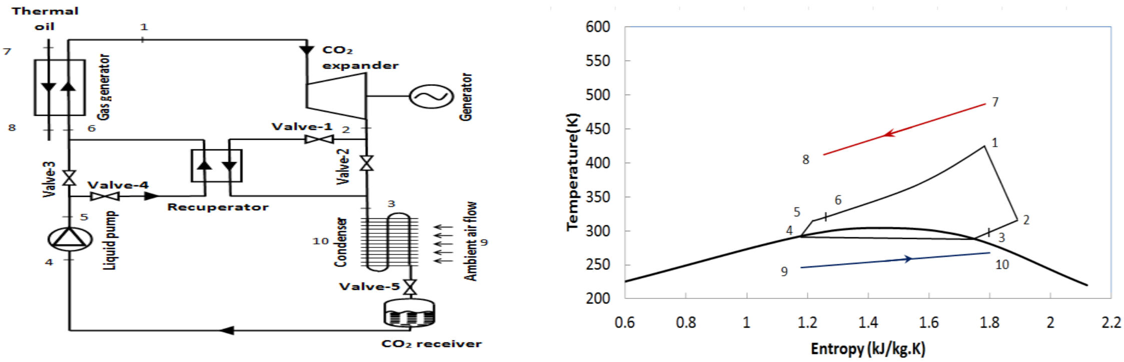
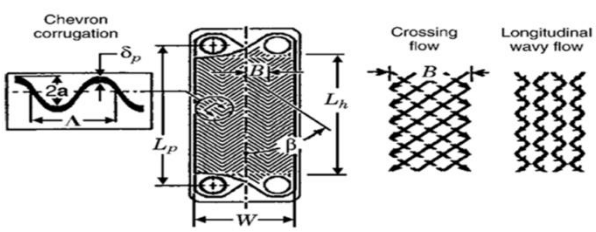


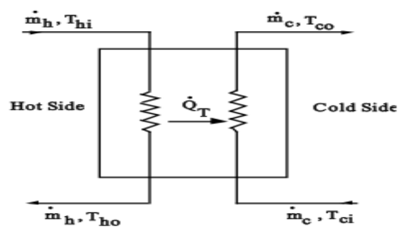
Fig.1. Schematic diagram of CO_2 power generation test rig and corresponding T-S diagram (with recuperator)

2.1 Models of CO_2 gas heater and recuperator

A plate CO_2 gas heater was installed in the T-CO_2 test rig with thermal oil and supercritical CO_2 respectively on hot and cold sides. In addition, a plate recuperator was added in the rig to evaluate its effect on system performance. As shown in Fig 1, the hot and cold sides of the recuperater are respectively low pressure CO_2 superheated gas from the turbine exit and high pressure subcooled liquid from the pump outlet. Since there is no phase change for both the CO_2 gas heater and recuperator, the same model can be applied to these two heat exchangers, as described below.



Geometrical parameters of a Chevron plate [6]



A counter flow mode

Fig. 2. Plate parameters and flow mode of the plate heat exchanger

As shown in Fig 2, internally, for each of the above plate heat exchangers, a number of Chevron plates are installed in parallel with fixed pitch between two neighbour plates. There are various requisite structural parameters describing the heat exchanger for model development. These include total number of plates (N), channel numbers per pass for both hot and cold fluids, distance between head plates, horizontal length of the plates (W), vertical length of the fluid path between the upper and lower ports (L_p), plate thickness (δ_p), the ratio of the developed length to the projected length (ϕ) and chevron angle (β) etc.

For the plate heat exchanger, as shown in Fig 2, counter flow profiles for hot and cold fluids are assumed and the effectiveness ε for the heat transfer is calculated:

$$\varepsilon = \frac{1-\exp\left(-\frac{UA}{C_{min}}\left(1-\frac{C_{min}}{C_{max}}\right)\right)}{1-\left(\frac{C_{min}}{C_{max}}\right)\exp\left(-\frac{UA}{C_{min}}\left(1-\frac{C_{min}}{C_{max}}\right)\right)} \tag{1}$$

where, the parameters in the right side of equation (1) can be calculated as below:

$$A=L_p\times W\times \Phi\times N \quad ; \quad U=\frac{1}{\frac{1}{\alpha_{hi}}+\frac{1}{\alpha_{ci}}} \quad ; \quad UA=U\times A \tag{2}$$

$$C_{hi}=\dot{m}_{hi}\, CP_{hi} \quad ; \quad C_{ci}=\dot{m}_{ci}\, CP_{ci} \tag{3}$$

$$C_{max}=\text{maximum value of } C_{hi} \text{ and } C_{ci} \tag{4}$$

$$C_{min}=\text{minimum value of } C_{hi} \text{ and } C_{ci} \tag{5}$$

The correlations from literature reference are used to calculate the fluid heat transfer coefficients of both hot and cold sides [6].

The heat capacity and fluid outlet temperatures of both hot and cold sides can therefore be calculated.

$$Q_T=\varepsilon C_{min}(T_{hi}-T_{ci}) \quad ; \quad T_{ho}=T_{hi}-\frac{Q_T}{C_{hi}} \quad ; \quad T_{co}=T_{ci}+\frac{Q_T}{C_{ci}} \tag{6}$$

2.2 Model validation

The developed gas heater model is therefore simulated at the same operating test conditions. It is then compared with corresponding test results including thermal oil outlet temperature and CO_2 outlet temperature. The comparison shows that thermal oil outlet temperatures are over-predicted by the model, while CO_2 outlet temperatures are under-predicted. One reason may be from the assumption of constant specific heat capacity on the CO_2 side which actually changes significantly during the supercritical heating process in the heat exchanger. However, the predicted temperature errors are mostly within 5K which are generally acceptable considering the high temperature ranges of both hot and cold fluids. Since all test results were based on the T-CO_2 power generation system without a recuperator, validation of the recuperator model could not be established. However, considering the similarity between recuperator and gas heater models, the recuperator model is expected to behave equivalently. It should be noted that the CO_2 finned-tube air cooled condenser model has been developed, validated and comprehensively explained by the authors before [7]. For the turbine and liquid pump models, conventional thermodynamic models were applied for the actual expansion and compression processes. These can therefore demonstrate an acceptable model of the T-CO_2 system.

3 Simulation of the T-CO_2 power generation system

The system model can be used to predict the effects of heat source and sink parameters on system performance. The operation conditions are specified in Table 1 for the system model simulation. In addition, power generation (W_{exp}) is controlled at 5 kW by changing either CO_2 pump speed or CO_2 flow rate. For each model simulation, only one parameter in the Table is changed while others remain constant.

Table 1. Operation conditions for model simulation

Thermal oil side (heat source)		Condenser side (heat sink)			Turbine inlet	Power generation
t_{oil} (^{0}C)	$\dot{m}_{oil}$(kg/s)	t_{air} (^{0}C)	ΔT_{sc} (K)	ΔT_{aircd}(K)	P_{expin} (bar)	W_{exp}(kW)
200~280	0.8~1.6	5~25	2	5	80~120	5

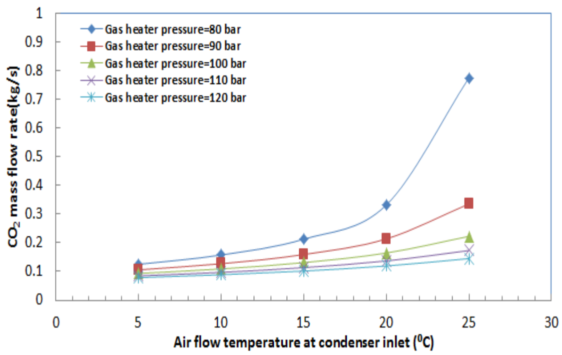


Fig. 3. Variation of CO_2 mass flow rate with different condenser air flow inlet temperature and CO_2 gas heater pressure

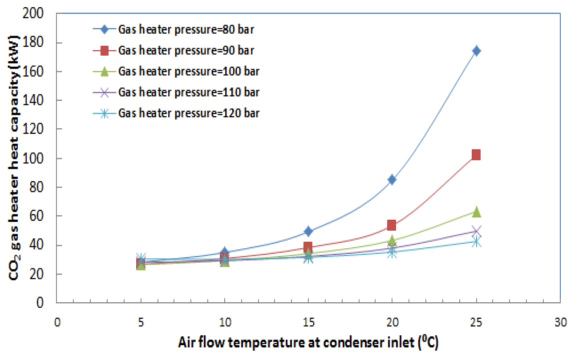


Fig. 4. Variation of CO_2 gas heater capacity with different condenser air flow inlet temperature and CO_2 gas heater pressure

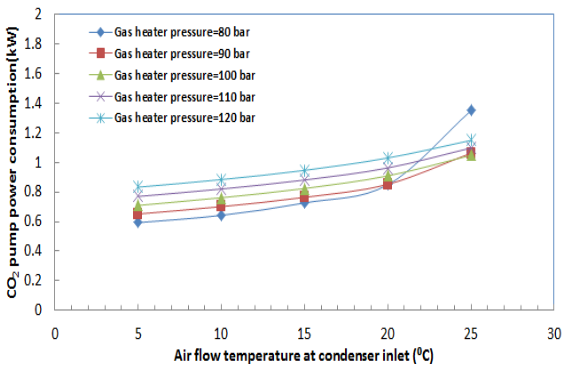


Fig. 5. Variation of CO_2 pump power consumption with different condenser air flow inlet temperature and CO_2 gas heater pressure

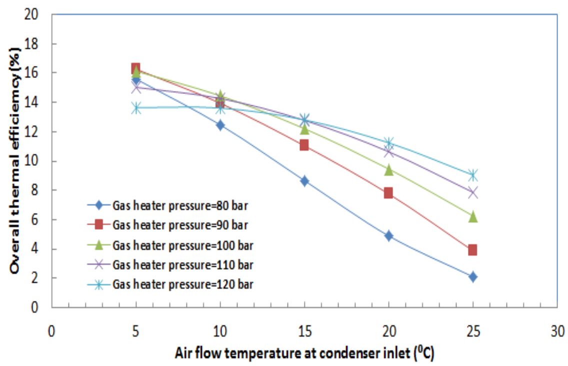


Fig. 6. Variation of overall thermal efficiency with different condenser air flow inlet temperature and CO_2 gas heater pressure

At different CO_2 turbine inlet pressures and condenser air flow inlet temperatures, the system performance parameters including CO_2 mass flow rate, gas heater heat capacity, pump power consumption and overall thermal efficiency can be predicted by the model and depicted in Figures 3-6 respectively. It is seen from Fig 3 that higher condenser air inlet temperatures will require larger CO_2 mass flow rates or pump speeds and such effects are significant when the CO_2 pressure at the turbine inlet decreases. The higher CO_2 mass flow rate also requires higher heat capacity from the gas heater, as shown in Fig 4 such that similar effects can present. The CO_2 pump power consumption is a function of both the CO_2 mass flow rate and pressure differences, as shown in Fig 5. Higher condenser air temperatures and CO_2 gas heater pressures can both increase pump power consumption. However, if the air reaches very high temperatures of above 25^0C, the CO_2 mass flow rate needs to be much higher to compensate, which will lead to a higher pump consumption even when the gas heater pressure is low. Consequently, the effect of the CO_2 gas heater pressure on overall system thermal efficiency is complex when the condenser air temperature is less than 15^0C, although its efficiency decreases with higher condenser air temperatures, as shown in Fig 6. At higher condenser air temperatures, efficiency increases at higher CO_2 gas heater pressures.

4. Conclusions

A transcritical CO_2 Rankine cycle (T-CO_2) is a prospective option for low-grade heat source power generation considering its natural working fluid properties and lower footprint as opposed to conventional ORC systems. A small scale test rig of T-CO_2 Rankine cycle was developed and investigations were carried out to investigate the effects of heat source and sink parameters on system performance. In addition, the components in the test system were modelled and integrated into an overall system model. The model was validated against test results and then used to further investigate the effects of gas heater pressure and heat sink parameters on the system performance at a larger scale. The simulation results show that both the CO_2 gas heater pressure and heat sink temperatures have a significant effect on system performance. There is an optimal gas heater pressure when the ambient air temperature is low. The system model can be an efficient and useful tool in the investigation of alternative system designs, performance evaluation and optimisation.

Acknowledgements

The authors would like to acknowledge the support received from GEA Searle and Research Councils UK (RCUK) for this research project

References

[1] Hung TC, Shai TY, Wang SK. A review of organic Rankine cycles (ORCs) for the recovery of low-grade waste heat, *Energy*1997; 22(7): 661-667.

[2] Handayani TP, Harvey AP, Reay DA. A review of organic Rankine cycle working fluid selection. *The Twelfth UK National Heat Transfer Conference2011*, Leeds.

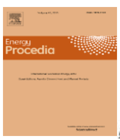
[3] Karellas S, Schuster A. Supercritical fluid parameters in organic Rankine cycle applications. *Int. J. of Thermodynamics* 2008; 11(3): 101-108.

[4] Zhang XR, Yamaguchi H, Uneno D, Fujima K, Enomoto M, Sawada N. Analysis of a novel solar energy-powered Rankine cycle for combined power and heat generation using supercritical carbon dioxide. *Renewable Energy* 2006;31:1839–1854.

[5] Chen Y, Lundqvist P, Johansson A, Platell P. A comparative study of the carbon dioxide transcritical power cycle compared with an organic rankine cycle with R123 as working fluid in waste heat recovery. *Applied Thermal Engineering* 2006; 26 : 2142–2147.

[6]Martin H. A theoretical approach to predict the performance of chevron-type plate heat exchangers. *Chem Eng Process* 1996; 35: 301–10.

[7] Ge YT, Tassou SA, Dewa Santosa I, Tsamos K. Design optimisation of CO_2 gas cooler/condenser in a refrigeration system. *Applied Energy* 2015;160:973-981.



Biography

Dr. Yunting Ge is Reader in the College of Engineering, Design and Physical Science at Brunel University London. Dr. Ge has 20+ years of research, application and development experience in refrigeration, energy conversion technologies and built environment controls and has published over 80 research papers in these fields.