Investigation on humidified gas turbine cycles with Maisotsenko-cycle-based air saturator

Guangya, Zhu; Chow, T. T.; Fong, K. F.; Lee, C. K.

Published in:
Energy Procedia

Published: 01/02/2019

Published version (DOI):
10.1016/j.egypro.2019.01.676

Publication details:
Guangya, Z., Chow, T. T., Fong, K. F., & Lee, C. K. (2019). Investigation on humidified gas turbine cycles with Maisotsenko-cycle-based air saturator. *Energy Procedia*, *158*, 5195-5200. https://doi.org/10.1016/j.egypro.2019.01.676

Available online at www.sciencedirect.com

ScienceDirect

Energy Procedia 158 (2019) 5195–5200

Energy Procedia

www.elsevier.com/locate/procedia

10th International Conference on Applied Energy (ICAE2018), 22-25 August 2018, Hong Kong, China

Investigation on humidified gas turbine cycles with Maisotsenko-cycle-based air saturator

Zhu Guangya[a,b], T. T. Chow[a] , K.F. Fong[a] and C. K. Lee[a]*

[a]*Division of Building Science and Technology, City University of Hong Kong, Hong Kong SAR, China*
[b]*Department of Architecture and Civil Engineering, City University of Hong Kong, Hong Kong SAR, China*

Abstract

The urge for more energy-efficient power plant systems triggered the design for more advanced power cycle designs. The current simple cycle gas turbine shows its advantage on cost but does not fully exploit the energy available in the flue gas. In this paper, two more efficient humidified gas turbine cycles were investigated. Type 1 was a hybrid design which combined an indirect evaporative cooler with a Maisotsenko cycle and Type 2 a conventional indirect evaporative cooler as air saturator. The simulation results indicated how the operating conditions such as pressure ratio and air flow changed with the designs. Both humidified gas turbine cycle systems offered better system efficiencies than a conventional gas turbine system with recuperator and the application of Maisotsenko cycle based air saturator could further improve the gas turbine cycle efficiency. Parametric studies were also conducted which highlighted the effects of water injection rate and part-load ratio on the comparative performances of the different humidified gas turbine cycle designs.

Keywords: Humidified gas turbine cycle, air saturator, indirect evaporator cooler, Maisotsenko cycle

1. Introduction

With the galloping growth of population and standard of modern life, the energy crisis becomes a serious challenge in our century. In response to the environmental concern, the application of more renewable technologies to enhance the energy supply is one solution. However, the low efficiency and the intermittent energy generation as

* Corresponding author. Tel.: +852 3442-5268; fax: +852 3442-9716.
E-mail address: chunklee@cityu.edu.hk

10.1016/j.egypro.2019.01.676

well as high initial cost and geographic requirement hinder the wide application of these kinds of technologies at present. Another more direct solution is to improve the efficiency of the power generation system [1].

Gas turbine application is important for the power generation industry. It usually consists of four parts: compressor, combustion chamber (CC), turbine and generator (G) and operates as a simple Brayton cycle. The traditional gas turbine power generation system efficiency was low at the early stage, but with the increasing of turbine inlet temperature and pressure ratio, its performance was improved [2]. Coupled with its lower capital cost, shorter start-up time, and lower maintains cost [3], gas turbine showed its advantage as power supply techniques. However, the simple gas turbine cycle efficiency remain inadequate to extent its application area. In order to exploit the gas turbine advantage, the humidified gas turbine cycle which utilizes an air-water mixture as the working fluid was proposed [4]. The humidified gas turbine cycle is a promising technology which not only improves the power generation of gas turbine but also reduces the NOx emission levels in the combustion chamber [5]. One of recently proposed humidified gas turbine cycles was the Maisotsenko combustion turbine cycle (MCTC) developed by a R&D firm based in Arvada, Colorado [6]. In the MCTC, the humidification tower, after cooler, recuperator, and economizer were combined into one component called air saturator which offered more economical advantages [7]. The flue gas could be cooled to a lower temperature by applying a high efficiency air saturator which was composed of a conventional indirect evaporator cooler at the top and a Maisotsenko cycle (M-cycle) at the bottom. Another claimed significant advantage of the MCTC was that the air temperature and humidity ratio at the air saturator outlet could be controlled by adjusting the quantity and location of water entering the shell side of the upper part. Currently, constant dew-point or wet-bulb efficiencies were employed in the simulation which might not reflect the real performance of MCTC under different conditions [7]. Therefore, a revised model for the MCTC was presented in this paper which took the dew-point and wet-bulb efficiencies variations into account to reveal its real potentials. Besides, MCTC with different air saturator designs were compared based on the same compressor, turbine and combustion chamber characteristics to test their influence on system efficiency. Finally, some critical parameters which affected the MCTC performance were studied.

Nomenclature

h	specific enthalpy (kJ/kg)
LHV	lower heating value (kJ/kg)
$\dot{m}$	mass flow rate (kg/s)
T	temperature (K)
W	electrical power output (kW)
ε	effectiveness
η	efficiency

Subscript

1..8	state point in the system
a	air
cc	combustion chamber
dew	dew-point temperature
f	fuel
net	overall system
w	water
wbt	wet-bulb temperature

2. System configuration

Simple gas turbine cycle with recuperator (SGTR) was used as reference system in this paper. Fig. 1a shows the layout of the SGTR. Compared to the simple Brayton cycle, a recuperator was added to recover part of the heat from the exhaust gas to pre-heat the compressed air before entering the combustion chamber. In this way, the thermal efficiency of the cycle could be increased. Fig. 1b depicts the layout of the MCTC according to Saghafifar and Gadalla [7]. The air saturator was employed in place of the recuperator in the SGTR. Compressed air was driven to the air saturator in which liquid water was added into the air. The preheated and humidified high-pressure air was then fed into the combustion chamber where it was mixed with the fuel for combustion. The generated high-temperature flue gas leaving the combustion chamber was utilized to drive the turbine for power generation. The low-pressure exhaust gas leaving the turbine was delivered back to the air saturator at the upper part to recover the exhaust energy for adding more moisture in the compressed air. To justify if this configuration (Type 1) was better, another design was also considered which solely employed a conventional indirect evaporator cooler as the air saturator (Type 2) as shown in Fig.1c. To derive the mathematical models for the various components in the system, the following assumptions were made:

- A constant polytropic efficiency and flow coefficient were assumed for compressor and turbine;
- The compression and expansion processes were adiabatic;
- All processes were assumed to be steady state;
- The air, fuel and exhaust gases were considered to be ideal gases;
- The combustion process was complete.

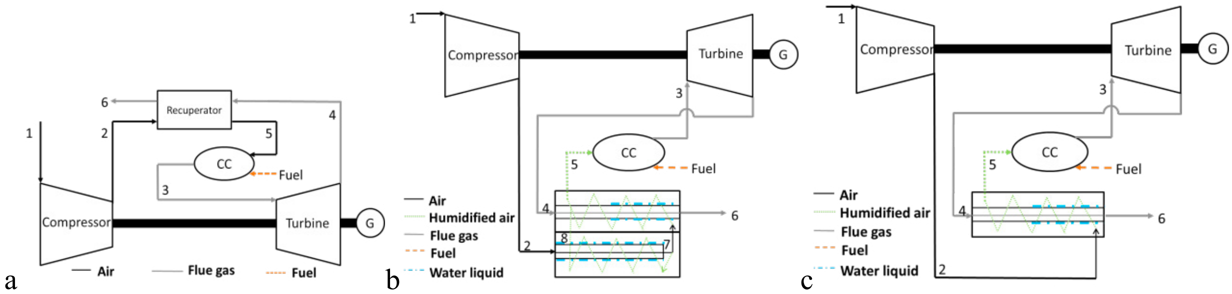

Fig. 1. Layout of the different gas turbine cycle (a. Simple gas turbine cycle with recuperator; b. Humidified gas turbine cycles with Type 1 air saturator; c. Humidified gas turbine cycles with Type 2 air saturator)

2.1. Compressor and Turbine

The mathematical model for calculating the compressor specific work, mass flow rate and compressor outlet air temperature could be found in Fong and Lee [8]. Due to the high turbine inlet temperature setting, turbine blade cooling was needed. In order to cool the blades, part of compressor outlet air was separated then mixed with the combustor exhaust gas in the first blade rows of the turbine. Similar to that in [7], it was assumed that the blade cooling air is mixed with the combustor's gases before entering the turbine.

2.2. Air saturator

For the Type 1 air saturator, the modelling method similar to that in [7] was employed. The lower part was a typical counter-flow regenerative heat and mass exchanger. The hot compressed air underwent sensible cooling from State 2 to State 7 (see Fig. 1b) with its performance represented by the dew-point effectiveness (ε_{dew}) as defined

$$\varepsilon_{dew} = \frac{T_2 - T_7}{T_2 - T_{dew,2}} \quad (1)$$

The upper part was similar to a typical indirect evaporative cooler. The temperature of the flue gas at State 6 could be estimated from

$$T_6 = T_4 - \varepsilon_{wbt}(T_4 - T_{wbt,7}) \quad (2)$$

where ε_{wbt} was the wet-bulb effectiveness of the upper channel. Type 2 air saturator was similar to the upper part of Type 1. The wet-bulb effectiveness was employed to determine the outlet temperature of the dry channel containing the flue gas so that

$$T_6 = T_4 - \varepsilon_{wbt}(T_4 - T_{wbt,2}) \quad (3)$$

As we mentioned before, the previous research [7] was based on the constant dew-point and wet-bulb efficiencies. However, the air saturator dew-point and wet-bulb effectiveness depended on its operation condition and configuration. To investigate, the dew-point effectiveness of the regenerative evaporative cooler as indicated in Fig 2a for the lower channel of Type 1 air saturator was calculated based on the same method adopted in another previous study [9]. The wet-bulb effectiveness employed for the Type 2 air saturator and the upper channel of Type 1 air saturator was based on the model by Wan *et al.* [10].

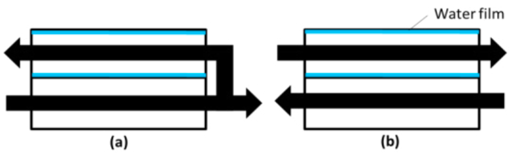

Fig. 2. Different evaporative cooler configurations (a. Regenerative evaporative cooler; b conventional indirect evaporative cooler)

2.3. Combustion chamber

The heat transfer across the combustion chamber can be calculated from

$$\dot{m}_{g,3}h_{g,3} = \eta_{cc}\dot{m}_f LHV_f + \dot{m}_{a,5}h_{a,5} \tag{4}$$

$$\dot{m}_{g,3} = \dot{m}_f + \dot{m}_{a,5} \tag{5}$$

Here, η_{cc} is the combustion chamber efficiency which takes into account the heat loss to the surrounding.

2.4. Overall performance

The overall system performance can be calculated from

$$\eta_{net} = \frac{W_{net}}{\dot{m}_f LHV_f} \tag{6}$$

Where W_{net} is the system electrical power output and η_{net} is the system overall efficiency.

3. Results and discussion

Table 1. Simulation results comparison for the various gas turbine systems at the reference condition.

Performance	Type 1	Type 2	SGTR
Compressor outlet temperature (K)	683.7	685.0	678.8
System outlet temperature (K)	488.8	625.8	706.1
Compressor outlet pressure (kPa)	1493	1503	1359
Turbine inlet pressure (kPa)	1348	1357	1275
Compressor mass flow rate (kg/s)	138.2	138.2	132.2
Turbine mass flow rate (kg/s)	163.1	165.8	134.8
Fuel mass flow rate (kg/s)	3.754	4.844	2.63
Compressor specific work input (kJ/kg)	407.0	408.5	395.1
Turbine specific work output (kJ/kg)	878.3	958.4	774.9
Dew-point effectiveness	0.700	N/A	N/A
Wet-bulb effectiveness	0.716	0.543	N/A
System efficiency (%)	44.43	40.54	39.26

Table 1 summarizes the simulation results for the various gas turbine systems at the reference inlet condition (ambient pressure taken as 100 kPa at 288 K and a humidity ratio of 0.0063 kg/kg) based on the same turbine inlet temperature of 1473 K. For the two humidified gas turbine systems, a fixed water injection rate of 20 kg/s was employed. It could be found that the efficiencies of the humidified gas turbine designs were higher than that of the SGTR. This highlighted the strength of air humidification in the enhancement of gas turbine performance. With different types of air saturators, the simulated compression ratios, fuel and turbine mass flow rates varied to some extents. Indeed, to reach the same turbine inlet temperature, the resulting fuel injection flow rates and compression ratios for the two humidified gas turbine systems were substantially higher than that of the SGTR. The calculated wet-bulb effectiveness with Type 1 air saturator was substantially higher than that with Type 2 air saturator. The main reason was that in the Type 1 air saturator, nearly 90% of the injected water was absorbed in the lower channel, i.e. the M-cycle. In other words, the water uptake by the indirect evaporative cooler in the Type 2 air saturator was much higher. This cohered with the finding from Wan *et al.* [10] that the wet-bulb effectiveness decreased with an increase in the water injection rate.

To assess the performances characteristic of humidified gas turbine systems, a parametric analysis was carried out to figure out the effect of water injection rate and part-load ratio on the relative performance of the two humidified gas turbine systems. In doing so, all other parameters were kept constant as those in the reference condition except the parameter being investigated.

3.1. Water injection rate

The major difference of humidified gas turbines over the conventional SGTR was the work fluid and it would be interesting to know how the water injection rate affected the system performance. Fig. 3 indicates the results for the different gas turbine systems design. With Type 1 air saturator, an optimal water injection rate around the reference value existed in which the system efficiency was maximized over the range of water injection rate analyzed. This could be explained by the fact that an increase in the water injection improved the mass flow to the turbine. However, it also reduced the air temperature entering the combustion chamber. With both opposing effects acted simultaneously, an optimal situation could be expected. The same should also hold with both Type 2 air saturator but was not found in Fig. 3. The reason was that the optimal point should likely appear at a very low water injection rate outside the range investigated. Nevertheless, the performance difference between the two types of humidified gas turbine systems generally reduced with the water injection rate particularly when it was lower than the optimal value for the Type 1 air saturator.

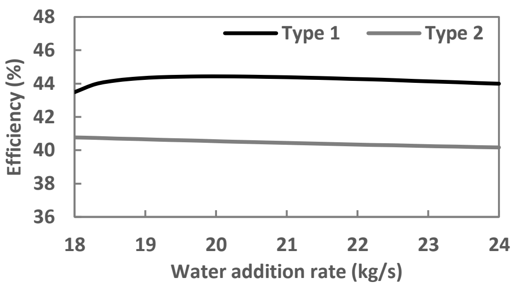

Fig.3. Variation of efficiency with water addition rate

3.2. Part-load ratio

Fig.4 shows the variation of the system efficiency with the part-load ratio of the gas turbine systems. Both humidified gas turbine systems performed better than the SGTR especially at a lower part-load ratio. Type 1 air saturator showed better performance throughout the whole study range of part-load ratio. Meanwhile, the performance discrepancy between Type 1 and Type 2 air saturators increased a little with the part-load ratio. Combined with the previous analysis results made, Type 1 air saturator was considered better.

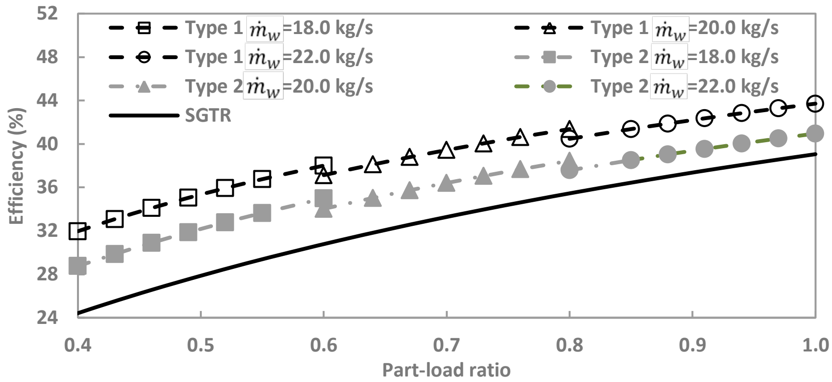

Fig.4. Variation of efficiency with part-load ratio

4. Conclusions

In this study, the impact of the air saturator performance on a humidified gas turbine cycle was investigated. Two types of air saturator designs were considered. Type 1 was a hybrid design which combined an indirect evaporative cooler with a Maisotsenko cycle, Type 2 being a conventional indirect evaporative cooler. Performance comparisons were made based on the adoption of the same compressor and turbine characteristics. The simulation results under the reference inlet air condition indicated that both types of humidified gas turbine systems offered higher efficiencies than that based on the conventional gas turbine system with recuperator, with the use of the more complicated Type 1 air saturator being better. To further reveal the comparative performances of the two types of humidified gas turbine cycles at different system water injection rate and part-load ratio, a parametric study was conducted. It was found that the performance differences decreased with a lower water injection rate. The impact of the part-load ratio was relatively mild. In view of the simulation results over the ranges of operating parameters investigated, it was concluded that the humidified gas turbine cycles with Maisotsenko-cycle-based air saturator was considered better for offering a higher power generation efficiency.

Acknowledgements

The work described in this article was financially supported by the research grants of the City University of Hong Kong (Strategic Research Project no. 7004804 and Contract Research Project no. 9231136)

References

[1] C.W. Chan, J. Ling-Chin, A.P. Roskilly, A review of chemical heat pumps, thermodynamic cycles and thermal energy storage technologies for low grade heat utilization, Applied Thermal Engineering 50 (2013) 1257-1273.
[2] C. Wu. Thermodynamic cycles: computer-aided design and optimization, CRC Press, Hoboken, 2003.
[3] S. AlFahed, F. Alasfour, H. Abdulrahim, The effect of elevated inlet air temperature and relative humidity on cogeneration system, International Journal of Energy Research 33(15) (2009) 1384–1394.
[4] M.H. Mahmood , M. Sultan, T. Miyazaki, S. Koyama, V.S. Maisotsenko, Overview of the Maisotsenko cycle – A way towards dew point evaporative cooling, Renewable and Sustainable Energy Reviews 66 (2016) 537–555.
[5] M. Jonsson, J. Yan. Humidified gas turbines review of proposed and implemented cycles, Energy 30(7) (2005) 1013–1078.
[6] V. Maisotsenko, L. Gillan, T. Heaton, A. Gillan, Power system and method, United States of America Patent US7007453 B2, 7 March, 2006. <http://www.google.com/patents/US7007453>.
[7] M. Saghafifar, M. Gadalla, Analysis of Maisotsenko open gas turbine power cycle with a detailed air saturator model, Applied Energy 149 (2015) 338-353.
[8] K.F. Fong, C.K. Lee, Dynamic performances of trigeneration systems using different prime movers for high-rise building application: A comparative study, Building Simulation 10 (2017) 509-523.
[9] G.Y. Zhu, T.T. Chow, C.K. Lee. Performance analysis of counter-flow regenerative heat and mass exchanger for indirect evaporative cooling based on data-driven model, Energy and Buildings 155 (2017) 503-512.
[10] Yangda Wan, Chengqin Ren, Zhao Wang, Yang Yang, Lei Yu. Numerical study and performance correlation development on counter- flow indirect evaporative air coolers, International Journal of Heat and Mass Transfer 115 (2017) 826–830.