Journal of Energy and Environmental Sustainability, 3 (2017) 51-55

An official publication of the International Society for Energy, Environment and Sustainability (ISEES)

Journal of Energy and Environmental Sustainability

Journal homepage : www.jees.in

Waste Heat recovery from Flue Gas Containing SO_2 Through CO_2 Based Combined Power and Refrigeration cycle

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ARTICLE INFO

Received : 01 May 2017
Revised : 29 May 2017
Accepted : 31 May 2017

Keywords:
Combined power and refrigeration; CO_2; NTU; waste heat recovery

ABSTRACT

CO_2 at supercritical state is a preferable working fluid for low grade waste heat recovery due to a low critical temperature of CO_2. In the present study a CO_2 based combined power and refrigeration cycle is proposed in which CO_2 coming out from the evaporator of the refrigeration system is pressurized to heat recovery unit pressure. The cycle is driven by the low grade waste heat of industrial flue gas containing small amount of SO_2.The basic objective of this study is to explore operating parameters for the best possible performance of the proposed combined power and refrigeration cycle driven by low grade waste heat. It is observed that refrigeration effect increases as a larger mass fraction of CO_2 enters the evaporator. However, this leads to a reduction in net cycle power output. Overall cycle performance (represented by 1st law efficiency) improves as larger mass fraction of CO_2 enters the evaporator. Exponential growth of the physical size (i.e. NTU) of the regenerator limits the maximum practical value of CO_2 mass flow into the evaporator.

Nomenclature:

h	Enthalpy in kJ/kg
c_{Pg}	Specific heat of flue gas in kJ/kgK
C_{min}	Minimum heat capacity in kJ/K
m_{CO2}	CO_2 flow rate in kg/s
m_g	Flue gas flow rate in kg/s
NTU_{REGEN}	Number of transfer unit for the regenerator
P_{COOLER}	Cooler pressure in MPa
Q_{RE}	Refrigeration effect in kW
r	Fraction of CO_2 mass entering into the evaporator
T_{gi}	Flue gas inlet temperature in K
T_{go}	Flue gas outlet temperature in K
W_{NET}	Cycle power output in kW
W_{REF}	Requisite power for producing Q_{RE} refrigeration effect by a vapor compression cycle in kW
η_t	1st law efficiency
1-10	state points

1. Introduction

As a substantial part of industrial energy input through combustion of fuel is finally rejected as low grade waste heat waste heat recovery would appear as one of the sustainable options in the energy sector. Secondary energy could be generated using industrial waste heat through transcritical and subcritical organic Rankine cycles (Wang et al. 2011; Wang et al. 2013; Hung et al. 1997; Yang 2016; Tian et al. 2016). CO_2 at supercritical state is preferred for low grade heat recover due to better matching of temperature of its temperature profile with that of heat source (Chen and Lundqvist 2011). Also CO_2 is having zero ODP and one GWP. Substantial studies were conducted to explore the capability of CO_2 based power cycle for generating power (Mondal and De 2015; 2015; 2017). Also CO_2 was utilized as the working fluid in refrigeration and heat pump cycle (Sarkar et al. 2005; Robinson et al. 1998). Wang et al. proposed a CO_2 based solar driven combined power and refrigeration cycle that utilized an ejector expansion device to couple a Brayton cycle with a transcritical refrigeration cycle (Wang et al. 2012). Wang et al. optimized performance of combined power and refrigeration cycles with CO_2 for utilization of IC engine exhaust (Wang et al. 2014).

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In the present study a transcritical CO_2 cycle is proposed that can produce simultaneous power output and refrigeration effect using low grade heat of industrial flue gas containing small amount of SO_2 In this study it is assumed that flue gas is available at 170⁰C. As flue gas temperature at the exit of the heat recovery unit (HRU) should be above 120⁰C (to avoid sulphuric acid condensation), very small amount of heat is available for heating in the HRU. In this situation higher thermal efficiency is the basic requirement for practical feasibility of any heat recovery scheme. Thus, in the present study CO_2stream coming out from the refrigerator evaporator is pressurised to heat recovery unit pressure and allowed to mix with the main CO_2 stream coming out from the regenerator. This improves the heat capacity ratio (ratio of heat capacity of hot fluid to that of cold fluid) of the regenerator and CO_2 enters the heat recovery unit at some higher temperature compared to a simple regenerative CO_2 power cycle. This will improve cycle performance as larger mass of CO_2can be heated by using by the specified mass of flue gas. Thus, proposing an environment-friendly and safe combined power and refrigeration cycle with a reasonable thermal efficiency and exploring the best possible operating condition for the proposed cycle through parametric analysis are the basic objectives of this study.

2. Methods

2.1 System description

Lay out and P-h diagram of the proposed cycle are presented in the Fig.1 (a) and Fig. 1(b) respectively. The CO_2 stream coming out from the turbine (i.e. state-2) after being cooled in the regenerator (i.e. process 2-3) and cooler (i.e. process 3-4) is split into two streams. One stream after undergoing a throttling process (i.e. process 4-5) enters the refrigerator evaporator. Another stream is pressurized to heat recovery unit pressure (i.e. state5) using a compressor. The CO_2 stream coming out from the evaporator at state-8 is pressurized to heat recovery unit pressure (i.e. state-9) and allowed to mix with the CO_2 stream coming out from the regenerator at state-6. After mixing the total CO_2 mass enters the heat recovery unit at state-10 and is heated to turbine inlet condition (i.e. state-1). The proposed cycle is a transcritical cycle as pressure of CO_2 in the heat recovery unit (HRU) is above the critical pressure and the corresponding value in the evaporator is below the critical pressure.

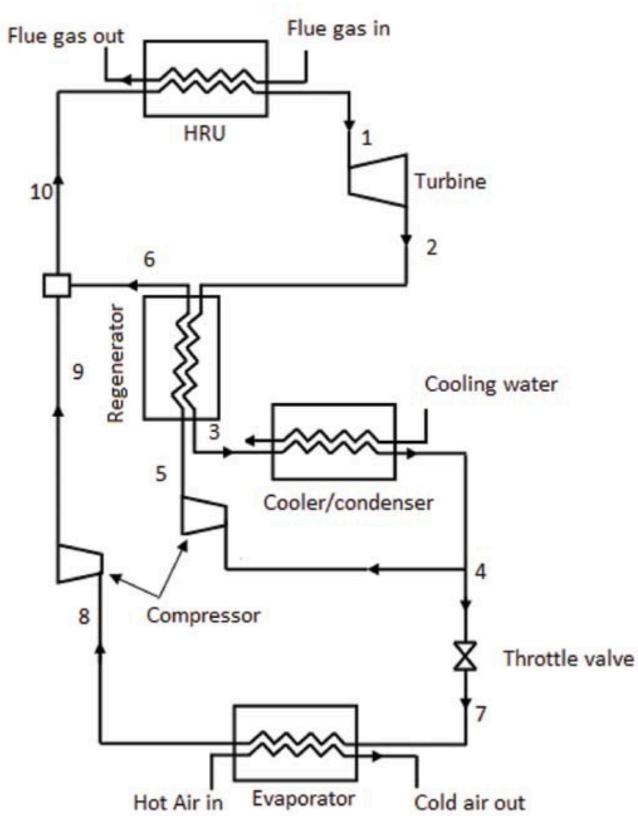

Fig. 1(a): Layout of the combined power and refrigeration cycle

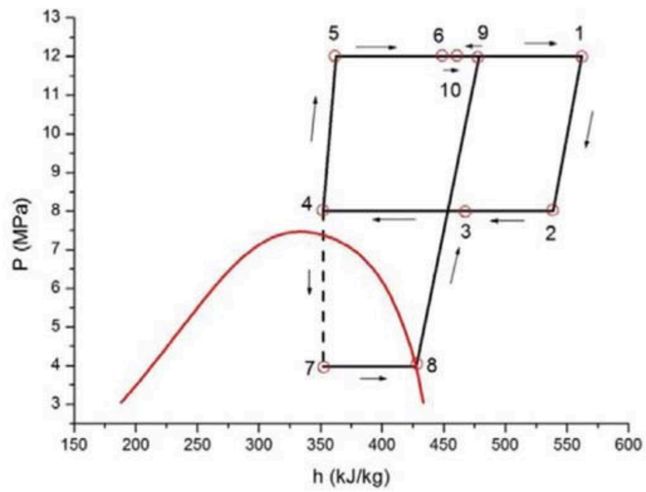

Fig. 1(b): P-h diagram of the combined power and refrigeration cycle

2.2. Mathematical Modelling

During the mathematical modelling it is assumed that 20 kg/s of flue gas is available at 170⁰C as the heat source. The flue gas exit temperature for gas heater is restricted to 120⁰C to avoid sulphuric acid condensation. Turbine and compressor isentropic efficiencies are 85% and 80% respectively. In the regenerator, CO_2 stream coming out from the turbine is cooled to lowest possible temperature assuming a 10⁰ C pinch point temperature difference. During entire modelling steady state operating condition is assumed. Pressure drop and extraneous heat loss are neglected.

From the energy balance of the heat recovery unit

$$m_{CO_2} = \frac{m_g c_{Pg}(T_{gi} - T_{go})}{h_1 - h_{10}} \tag{1}$$

Now cycle power output will be

$$W_{NET} = m_{CO_2}(h_1 - h_2) - m_{CO_2}(1-r)(h_5 - h_4) - m_{CO_2} r(h_9 - h_8) \tag{2}$$

Refrigeration effect can be calculated by equation-(3) as follows:

$$Q_{RE} = m_{CO_2} r(h_8 - h_7) \tag{3}$$

Now 1st law efficiency will be

$$\eta_I = \frac{W_{NET} + W_{REF}}{m_g c_{Pg}(T_{gi} - T_{go})} \tag{4}$$

In equation (4), W_{REF} is the power input would have been required to run a vapour compression refrigerator between the evaporator and cooler pressure for obtaining Q_{RE} refrigeration effect.

The physical size of the refrigerator is denoted by NTU and this can be determined as follows:

$$NTU_{REGEN} = \sum_{i=1}^{20} NTU_i \tag{5}$$

For determining NTU of regenerator entire regenerator is divided in 20 elements. In the above equation NTU_i is the NTU of ith element.

3. Results and Discussion

In the present study a CO_2 based combined power and refrigeration cycle is proposed. Assuming 150⁰C turbine inlet temperature and 12MPa turbine inlet pressure, effects varying cooler pressure and CO_2 mass fraction flow to evaporator on cycle performance are evaluated. During the analysis evaporator temperature is assumed to be fixed at 5⁰C and corresponding CO_2 exit temperature form cooler be 35⁰C.

Effect of varying cooler pressure on net cycle power output is shown in Fig.2 (a). It is observed that there exists an optimum cooler pressure for a maximum cycle power output. Both turbine power output and power input to the compressor operating between cooler and regenerator decrease with an increase in cooler pressure. This can be better explained by Fig. 2 (b). It is observed in Fig. 2(b) that initially the slope of the compression line (i.e. 4-5) sharply decreases with a small increase in cooler pressure. This is occurring due to very large shift of this compression line towards the left of the P-h diagram along the 35^{0}C isotherm. Larger slope of the compression line in the P-h diagram represents smaller enthalpy change or power input during the process of compression. Thus, initially with an increase in cooler pressure power input to the compressor operating between the cooler and the regenerator decreases at a faster rate compared to that of the turbine power output. This results in larger cycle power output. However, it is evident from the Fig. 2 (b) that this shifting of the compression line towards the left of the P-h diagram with varying cooler pressure becomes almost negligible above a certain value of cooler pressure. Above this pressure variation in slope of the compression line (hence, the power input to compressor operating between the cooler and the regenerator) becomes insensitive to varying cooler pressure. But turbine power output decreases almost at the same rate as mentioned earlier. Thus, net cycle power output decreases with any further increase in cooler pressure above this particular value of cooler pressure.

From Fig. 2 (a) it is also clear that cycle power output sharply decreases as larger fraction of CO_2 mass enters into the evaporator due to larger power consumption of the compressor operating between evaporator and gas heater pressures. Above certain value of CO_2 mass fraction entering the evaporator cycle power output even becomes negative or cycle becomes merely a refrigeration cycle.

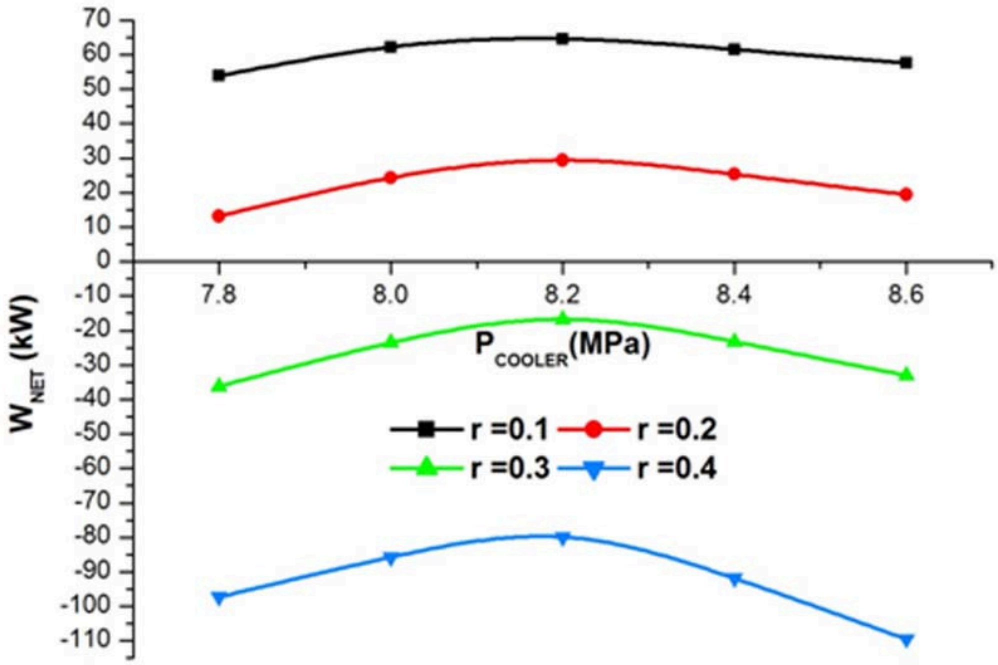

Fig. 2(a): Effect of varying cooler pressure on cycle power output

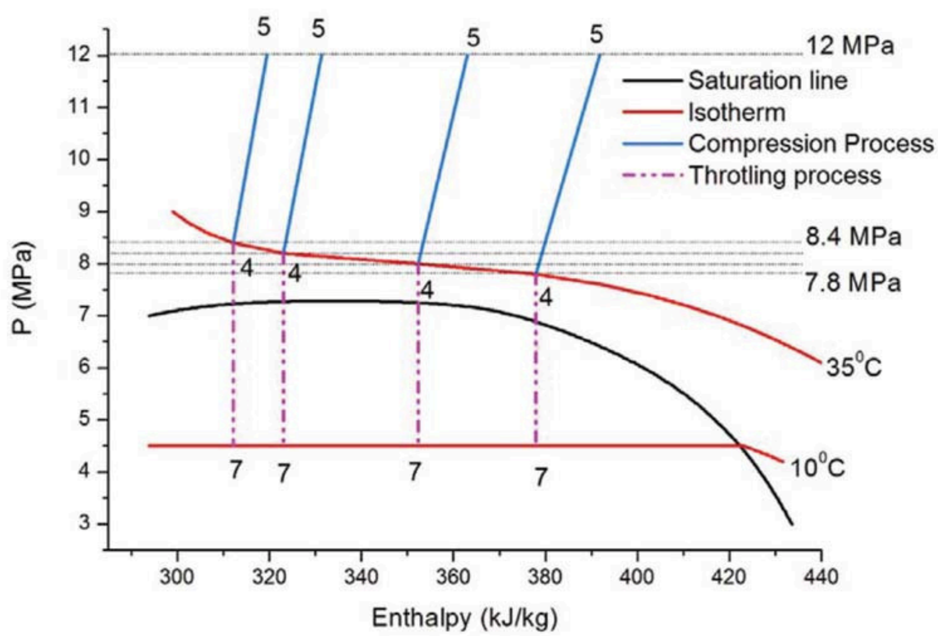

Fig. 2(b): P-h diagram showing shifting of state points associated with compression and throttling process for varying P_{COOLER}

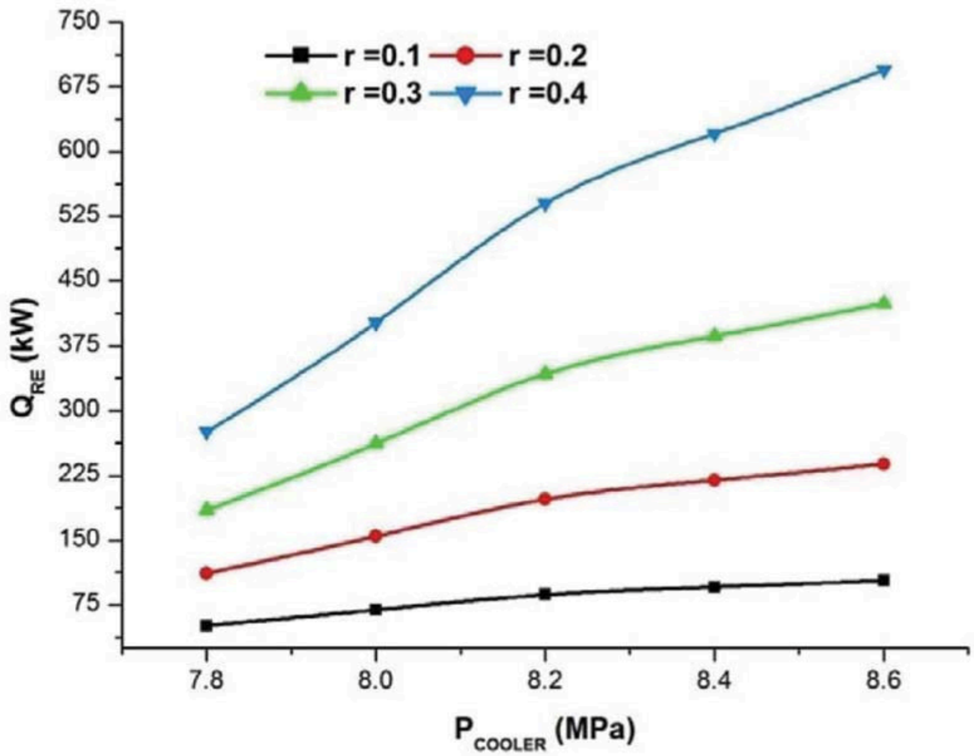

Fig. 3: Effect of varying cooler pressure on refrigeration effect.

It is observed in Fig.3 that for specified refrigerant flow to the evaporator, refrigeration effect increases with an increase in cooler pressure. This is because specific enthalpy of CO_2 stream at evaporator inlet or cooler exit decreases due to shifting of the state point-4 towards the left of the p-h diagram along the 35^{0}C isotherm as shown in Fig. 2 (b). Above a certain cooler pressure improvement in refrigeration effect is not very sharp due to smaller shifting of state point-4 towards the left of the p-h diagram along the 35^{0}C isotherm with an elevated value of cooler pressure. It is clear from Fig.3 that refrigeration effect increases as larger mass fraction of CO_2 enters the evaporator.

It is observed in Fig.4 that there exists an optimum cooler pressure for the maximum 1st law efficiency for a specified flow rate of CO2 entering into the evaporator. Initially 1st law efficiency increases with an increase in cooler pressure as both cycle power output and refrigeration effect increase.However, cycle power output starts to decrease beyond a certain value of cooler pressure. Beyond the optimum value of cooler pressure decreasing trend of cycle power output becomes the dominating factor. Also it is observed that optimum value of 1st law efficiency improves as larger fraction of CO_2 mass enters the evaporator. With larger mass fraction of CO_2 entering into the evaporator, not only refrigeration effect improves but also larger mass of CO_2 can be heated by using same heat input in the heat recovery unit as CO_2 enters the heat recovery unit at some higher temperature. The optimum cooler pressure corresponding to maximum 1st law efficiency also increases as larger mass of CO_2 enters the evaporator. This is due to corresponding improvement of refrigeration effect associated with the larger mass flow of CO_2 in to the evaporator.

It is observed in FIG.5 that an increment in CO_2 mass fraction flow to the evaporator leads to a larger size of regenerator (specified by NTU). It should be noted that the specific heat of the CO_2 stream coming out from the regenerator at state-5 is appreciably large compared to exhaust CO_2 stream coming out from the turbine and entering to the regenerator as hotter stream at state-2 (Refer to Fig. 1 (b)). This smaller mass fraction flow of CO_2 to the evaporator results in a smaller heat capacity ratio and higher effective temperature difference between hot and cold streams of the regenerator. The corresponding size of the regenerator (specified by NTU) is considerably small. However, as a larger fraction of CO_2 mass flows into the evaporator heat capacity of the CO_2 stream at the exit of the compressor (i.e. state-5) decreases. This results in larger heat capacity ratio of the regenerator and smaller effective temperature difference between hot and cold streams of CO_2. Due to this smaller effective temperature difference size of the regenerator increases with an increase in CO_2 mass fraction flow into the regenerator.

For specified cooler pressure above certain value of CO_2 mass fraction flow to the evaporator this increment in NTU of the regenerator is very rapid. Thus CO_2 mass fraction flowing into the evaporator should not be increased above certain value for maintaining an economically feasible size of the regenerator.

It is evident from Fig.5 that NTU of the regenerator increases with an increment in cooler pressure due to larger heat duty of the regenerator. Heat duty increases due to larger regenerator inlet temperature of the exhaust CO_2 stream coming out from the turbine. Also with increasing cooler pressure this stream can be cooled to lower temperature as regenerator inlet temperature of colder CO_2 stream decreases.

Obviously effective temperature difference increases between hot and cold fluids with an elevated cooler pressure. However this effect is becomes less effective with larger cooler pressure. Inlet temperature of colder fluid into the regenerator becomes almost constant for larger cooler pressures. So exit temperature of this colder CO_2 stream comes closer to the exhaust CO_2 stream by absorbing larger amount of heat released by hotter CO_2 stream during the regeneration process at larger cooler pressure. Thus regenerator size increases at a slower rate with an initial increment in cooler pressure. However, this increment becomes faster as cooler pressure increases due to smaller increment in effective temperature difference between hot and cold streams.

4. Conclusions

In the present study a CO_2 based combined power and refrigeration cycle driven by low grade heat of flue gas is proposed. It is observed that

- Cycle power output decreases as larger fraction of CO_2 mass enters the evaporator. However, this results in larger refrigeration effect.
- Cycle performance is represented by the 1st law efficiency to consider combined effect of the cycle power output and the refrigeration effect. It is observed that there exists an optimum cooler pressure for maximum 1st law efficiency for specified mass fraction of CO_2 entering to the evaporator.
- The optimum value of 1stlaw efficiency increases as larger mass fraction of CO_2 enters into the evaporator.
- However, the mass fraction of CO_2 entering to the evaporator should not be increased above a certain value to maintain a practically feasible size of the regenerator.

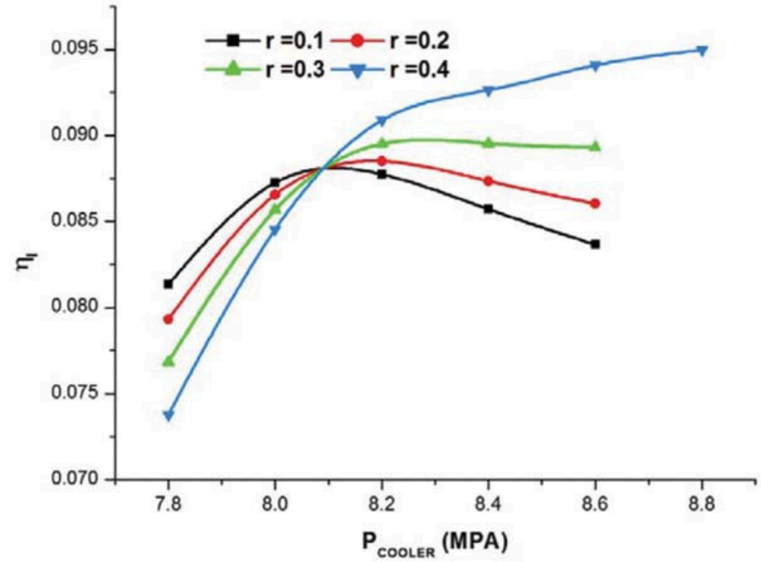

Fig. 4: Effect of varying cooler pressure on 1st law efficiency

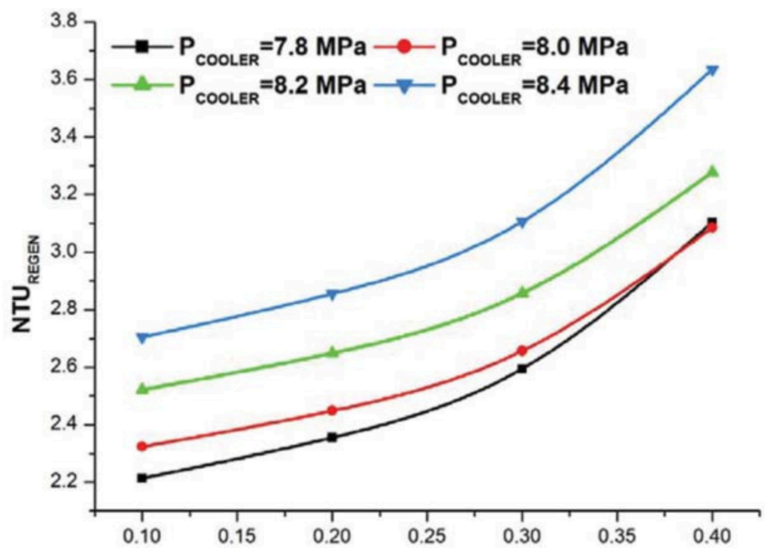

Fig.5: Effect of varying CO_2 mass fraction flow into the evaporator on NTU of the regenerator.

References

1. Chen Y, and Lundqvist P, 2011, The CO_2 power cycle for low grade heat recovery: Discussion on temperature profiles in system heat exchangers, ASME Power Conference 1. 385-392.
2. Hung T.C, Shai T.Y, Wang S.K, 1997, A review of organic Rankine cycles (ORCs) for the recovery of the low-grade waste heat, Energy, 22. 661-667.
3. Jiangfeng Wang, Pan Zhao, Xiaoqiang Niu, Yiping Dai, 2012, Parametric analysis of a new combined cooling, heating and power system with transcritical CO_2 driven by solar energy, Applied Energy, 94, 58-64
4. Mondal S, De S, 2015, CO_2 based Power cycle with multi-stage compression and intercooling for low temperature waste heat recovery, Energy, 90, 1132-1143.
5. Mondal S, De S, 2015, Transcritical CO_2 power cycle – effects of regenerative heating using turbine bleed gas at intermediate pressure, Energy, 87, 95-103.
6. Mondal S, De S, 2017, Power by waste heat recovery from low temperature industrial flue gas by Organic Flash Cycle (OFC) and Transcritical-CO2 power cycle: a comparative study through combined thermodynamic and economic analysis, Energy, Energy, 121, 832-84.
7. Robinson Douglas M., Groll Eckhard A, 1998, Efficiencies of transcritical CO_2 cycles with and without an expansion turbine: Rendement de cycles transcritiques au CO_2 avec et sans turbine d'expansion, International Journal of Refrigeration, 21, 577-589.
8. Sarkar J, Bhattacharyya Souvik, Ram Gopal M, 2005, Transcritical CO_2 heat pump systems:exergy analysis including heattransfer and fluid flow effects, Energy conversion and management, 46, 2053-2067.
9. Tian Ran, An Qingsong, Zhai Huixing, Shi Lin, 2016,Performance analyses of transcritical organic Rankine cycles with large variations of the thermophysical properties in the pseudocritical region, Applied thermal engineering, 101.183-190.
10. Wang E.H, Zhang H.G, Fan B.Y, Ouyang M.G, Y. Zhao, Mu Q.H., 2011, Study of working fluid selection of organic Rankine cycle (ORC) for engine waste heat recovery, Energy, 36, 3406-3418.
11. Wang Jiangfeng, Wang Zhequan, Wang Man, Ma Shaolin, DaiYiping, 2013, Thermodynamic analysis and optimization of an (organic Rankine cycle) ORC using low grade heat source, Energy, 49. 356-365.
12. Wang S, Bai K, Xie Y, Di J, Cheng S, 2014, Analysis of Combined Power and Refrigeration GenerationUsing the Carbon Dioxide Thermodynamic Cycle to Recover the Waste Heat of an Internal Combustion Engine, Mathematical Problems in Engineering, http://dx.doi.org/10.1155/2014/68939.
13. Yang Min-Hsiung, 2016,Optimizations of the waste heat recovery system for a large marine diesel engine based on transcritical Rankine cycle, Energy, 113. 1109-1124