Energy 81 (2015) 344–351

Contents lists available at ScienceDirect

Energy

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

Analyses and optimization of a supercritical N_2O Rankine cycle for low-grade heat conversion

Jahar Sarkar*

Department of Mechanical Engineering, Indian Institute of Technology (B.H.U.), Varanasi, UP 221005, India

ARTICLE INFO

Article history:
Received 9 July 2014
Received in revised form
24 October 2014
Accepted 17 December 2014
Available online 13 January 2015

Keywords:
Supercritical Rankine cycle
Working fluids
Nitrous oxide
Exergetic performance
Component compactness
Optimization

ABSTRACT

Analyses and operating pressure optimization of the supercritical Rankine cycles with and without regenerator and reheating for low-grade heat conversion have been conducted using N_2O as a working fluid and compared with its counterpart CO_2 based on various performance indicators. N_2O is better in terms of net power output, thermal efficiency and exergetic efficiency and N_2O works at much lower pressures at optimum operation; whereas, CO_2 is advantageous in terms of turbine size, expansion ratio and heat transfer requirement. The choice of optimum operating conditions will differ depending on the chosen performance indicator. Hence, there is a need of trade-off between various indicators. Component wise irreversibility distribution shows the similar trends for both working fluids. With the increase in cycle temperature lift, both turbine shape parameter and heat transfer requirement decrease, leading to more compactness. Higher pump and turbine isentropic efficiencies yield higher optimum turbine inlet pressure, and lower heat transfer requirement and turbine size. Uses of internal heat exchanger and reheating in the supercritical Rankine cycle not only improve the performances, it also constitutes an excellent compromise between various performance indicators based optimizations. Present study reveals that N_2O is a potential option for the supercritical Rankine cycle.

1. Introduction

Due to the efficient temperature matching between the heat source and the working fluid, the supercritical Rankine cycle has considerable potential as a mean for converting low-grade heat (geothermal heat, industrial and mobile waste heat, solar energy, etc.) into electricity, however, it has not received as much attention as the organic Rankine cycle [1]. As a working fluid for supercritical Rankine cycle, carbon dioxide has various advantages such as nontoxic, inflammable, easily available, well known properties, low critical point, relative inertness, stability, compatible with standard materials and lubricants, component compactness, favorable thermodynamics and transports properties, little environmental impact and low cost [1–3]. Within the last decade, a large number of researches have been done for supercritical CO_2 Rankine cycle, which includes energetic, exergetic, heat transfer and cost analyses and optimization [4–6], comparison with other working fluids or cycles [7–11], modified cycle with internal heat exchanger and reheating [11,12] and use of waste heat, geothermal heat and solar energy as sources [1,13].

Nitrous oxide is the another natural working fluid, which has very similar molecular weight, critical pressure and temperature cause nearly the similar behavior to CO_2 with respect to system temperature and pressure, properties and compactness. Nitrous oxide is widely used in supercritical fluid extractions [14], but it remains largely unexplored as a working fluid in heat engine and heat pump cycles. It has been proposed as refrigerant for transcritical refrigeration and heat pump cycles as well as low temperature fluid in cascade refrigeration system and some theoretical studies have been reported [15–19]. Recently, Sarkar [20] has proposed nitrous oxide as a working fluid in supercritical recompression Brayton power cycle and conducted thermodynamic analyses and optimization studies. However, to the best of the authors' knowledge, no study has been reported on Rankine cycle using nitrous oxide.

The main objective of the present exercise is to use of N_2O in the supercritical Rankine cycle as a working fluid for low temperature heat source. The theoretical analysis and optimization of cycle pressure ratio have been performed to study the effects of various design and operating conditions on key parameters such as net

* Corresponding author. Tel.: +91 991 9787557; fax: +91 542 2368428.
E-mail address: jsarkar.mec@itbhu.ac.in.

http://dx.doi.org/10.1016/j.energy.2014.12.047

power output, energetic and exergetic efficiencies, and component sizes. Effects of using internal heat exchanger and reheating on the cycle performance have been also studied. Finally, various key parameters based comparison with its counterpart CO_2 and other working fluid in supercritical Rankine cycle is presented as well.

2. Theoretical modeling and simulation

In the supercritical Rankine cycle, heat addition and heat rejection occur at supercritical pressure and subcritical pressure, respectively. The layout and corresponding T-s diagram of a supercritical N_2O Rankine cycle are shown in Fig. 1. As shown, a stream of the saturated N_2O liquid is pumped above its critical pressure (1-2), and then heated isobarically from liquid directly to supercritical vapor (2-3); the supercritical vapor is expanded in the turbine to extract mechanical work (3-4); after expansion, the fluid is desuperheated and condensed in the condenser by dissipating heat to a heat sink (4-1); the condensed liquid is then pumped to the high pressure again, which completes the cycle. As the pump handles only liquid, the pump outlet temperature (state 2) should be less than critical temperature for proper operation.

The supercritical Rankine cycle has been modeled based on the energy and exergy balances of individual components to yield the equations that follow. The following general assumptions have been made for the analysis:

- Each component is considered as a steady-state steady-flow system
- Pressure drop in all the heat exchangers is negligibly small
- Changes in kinetic and potential energies in each component are negligible
- Expansion and compression processes have given isentropic efficiencies
- Heat transfer with the ambient is negligible except condenser
- Saturated liquid is supposed at the condenser exit

The pump and turbine work transfer rates are given by, respectively,

$$\dot{W}_p = \dot{m}(h_2 - h_1) \tag{1}$$

$$\dot{W}_t = \dot{m}(h_3 - h_4) \tag{2}$$

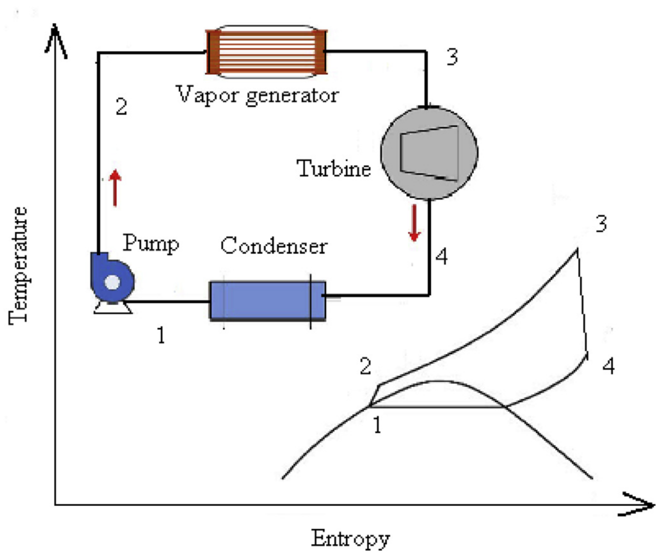

Fig. 1. Layout and T-s diagram of supercritical N_2O Rankine cycle.

The heat input and heat rejection rates are given by, respectively,

$$\dot{Q}_h = \dot{m}(h_3 - h_2) \tag{3}$$

$$\dot{Q}_c = \dot{m}(h_4 - h_1) \tag{4}$$

According to the first law of thermodynamics, the net power output is given by,

$$\dot{W}_{net} = \dot{W}_t - \dot{W}_p = \dot{Q}_h - \dot{Q}_c \tag{5}$$

Finally, the thermal efficiency of cycle is given by,

$$\eta_{th} = \dot{W}_{net} \big/ \dot{Q}_h \tag{6}$$

The turbine volumetric expansion ratio, which is used to measure of turbine size and staging, is given by,

$$Exp_r = v_4/v_3 \tag{7}$$

Furthermore, the turbine size parameter is given by Ref. [2],

$$SP = \sqrt{\dot{m}v_4} \Big/ \sqrt[4]{(h_3 - h_4)} \tag{8}$$

Similar to the volumetric cooling capacity used in refrigeration cycle to measure compressor compactness [17], the new parameter, volumetric net power output for Rankine cycle (to measure turbine compactness) can be defined as,

$$V_w = \dot{W}_{net} \big/ (\dot{m}v_4) \tag{9}$$

The objective of the finite size thermodynamic optimization is to determine the operating conditions which minimise the total UA of two heat exchangers (gas generator and condenser). This parameter is directly related to the heat exchanger surfaces and is often used to give a global idea of their dimensions. Assuming the constant temperature heat source and heat sink, the total heat transfer requirement, which can be used to measure compactness of heat exchangers, can be approximated as:

$$UA = \frac{\dot{Q}_h}{T_s - T_{m,h}} + \frac{\dot{Q}_c}{T_{m,c} - T_0} \tag{10}$$

where, $T_{m,h}$ and $T_{m,c}$ are mean temperatures of heat addition and rejection, respectively, which are given by,

$$T_{m,h} = (h_3 - h_2)/(s_3 - s_2) \tag{11}$$

$$T_{m,c} = (h_4 - h_1)/(s_4 - s_1) \tag{12}$$

Now, combining Eqs. (5), (6) and (10), the total heat transfer requirement per unit net power output can be expressed as:

$$\frac{UA}{\dot{W}_{net}} = \frac{1/\eta_{th}}{T_s - T_{m,h}} + \frac{(1/\eta_{th}) - 1}{T_{m,c} - T_0} \tag{13}$$

Actually, the effective temperature difference for heat transfer depends on many factors and this issue is more complicated in heat addition for present cycle due to sharp variation of the thermophysical properties. Hence, the actual data may differ with the predicted value from simplified Eq. (13). However, this will give the relative measure with other working fluids.

An exergetic analysis is necessary to know the extent of irreversibility in each process, identify where irreversibility happens, and therefore the potential of improvements. The related