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Exergoeconomic analysis of carbon dioxide transcritical refrigeration machines

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ABSTRACT

In the last two decades many scientific papers and reports have been published in the field of the application of the carbon dioxide as a refrigerant for refrigeration (heat pump) systems. A simple transcritical CO_2 refrigeration machine is evaluated from the perspectives of energetic, exergetic, economic and exergoeconomic analyses. Special attention has been paid to the transcritical cycle under hot climatic conditions. The main goal of this paper is to define the energy and cost efficient transcritical CO_2 refrigeration machine, therefore the options for the structure and parametric improvements are discussed. Introducing the economizer as an auxiliary component for one-stage transcritical CO_2 refrigeration machine helps us to decrease the total cost of the final product by approximately 14%.

Analyse Exergoéconomique de machines frigorifiques transcritiques au dioxyde de carbone

Mots clés : Machine frigorifique transcritique ; Dioxyde de carbone ; Analyse énergétique ; Analyse exergétique ; Analyse économique ; Analyse exergoéconomique

1. Introduction

The application of carbon dioxide (CO_2) for refrigeration is well known, including operations based on a transcritical cycle. However, only from the 1990's the refrigeration machines using CO_2 as a refrigerant have been in the focus of researchers and engineers. The reason is that the interest to the so-called "natural refrigerants" (carbon dioxide, ammonia, propane, butane, and water) is renewed, especially for CO_2, due to considerations related to Ozone Depletion Potential

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Nomenclature

A	surface area [m^2]
$\dot{C}$	cost rate associated with an exergy stream [€ $(h)^{-1}$]
c	cost per unit of exergy [€ $(GJ)^{-1}$]
$\bar{c}$	cost per unit of energy [€ $(GJ)^{-1}$]
COP	coefficient of performance [–]
h	specific enthalpy [kJ $(kg)^{-1}$]
$\dot{E}$	exergy rate [W]
e	specific exergy [kJ $(kg)^{-1}$]
$\dot{m}$	mass flow rate [kg $(s)^{-1}$]
p	pressure [Pa]
$\dot{Q}$	heat rate [W]
s	specific entropy [kJ $(kg\ K)^{-1}$]
T	temperature [°C]
$\dot{W}$	power [W]
$\dot{Z}$	cost rate associated with investment expenditures [€ $(h)^{-1}$]
0	reference state for the exergy analysis

Greek symbols

ε	exergy efficiency [–]
η	isentropic efficiency [–]

Abbreviations

CM	compressor
EC	economizer
EM	electrical motor
EV	evaporator
EXV	expansion valve
GC	gas cooler

Subs- and superscripts

a	average
cm	cooling medium
D	exergy destruction
F	fuel
k	k-th component
P	exergy of product
sr	secondary refrigerant
tot	total

(ODP) and Global Warming Potential (GWP), which have restricted the use of CFC's and HFC's as refrigerants (Montreal protocol, 1987). CO_2 has some unique properties that make this refrigerant completely different than other "natural refrigerants". The technical developments during the last decades helped to overcome many of the barriers for the wide application of CO_2, but still we need to investigate the rational application of this refrigerant.

A large number of scientific publications related to the theoretical and practical investigations of the different refrigeration and heat pump systems using CO_2 followed after publication of the papers published by Lorentzen and Pettersen (1993), Lorentzen (1994). Several papers that have already been published in the decade of 2000 (15th Informatory Note on Refrigerants, 2000; Kim et al., 2004; Pearson, 2005), contain excellent reviews of many publications that have been reported. However, in all these publications only Northern European climatic conditions are considered for the operation of refrigeration and heat-pump systems.

The Middle East is an interesting region of the world to study, because this region has experienced impressive increases in economic growth, and energy demand (Sadorsky, 2011). For countries with hot climates, the energy consumption related to refrigeration processes is much higher than for other countries. It is caused by the expanded application of refrigeration processes (especially for air-conditioning systems) and by the higher temperature of the environment (temperature of a cooling medium) that leads to a higher pressure ratio within the refrigeration system, and, therefore to higher energy consumption.

The operation of a CO_2 refrigeration machine at a high temperature of the environment can be similar to the operation of a CO_2 heat pump. Therefore, the following publications with corresponding assumptions and results have been considered here: Neksfit et al. (1998) reported optimal values of the pressure as well as the isentropic and volumetric efficiencies of the CO_2 compressors for heat pump applications in the range of the inlet water temperature between 7 and 20 °C and corresponding temperature of the evaporation between −25 °C and 15 °C, as well as hot water temperature between 55 and 80 °C. In this range of temperatures, the pressure ratio is varied between 2 and 5. The isentropic efficiency of the compressor is varied between 0.81 and 0.75 and the volumetric efficiency between 0.9 and 0.78. Schmidtt et al. (1998) investigated the characteristics of high-temperature heat pumps with a transcritical CO_2 process for drying purposes with a maximal temperature of 60 °C. The isentropic efficiency of the compressor was varied between 0.65 and 0.7.

An interesting review of CO_2-based heat-pump systems is published by Neksa (2002), however, only a relative low temperature for the hot water is considered with a maximal pressure of 90 bar and a minimal pressure of 35 bar. For these operation conditions, the isentropic efficiency of the compressor was 0.92 for the pressure ratio 2.4 and 0.68 for the pressure ratio 3.2. Cecchinato et al. (2005) reported a similar heat-pump system with maximal temperature of the hot water of 45 °C. The maximal pressure within the gas cooler is assumed to be 115 bar, whereas the isentropic efficiency of the compressor varies between 0.6 and 0.63.

The effect of ambient temperatures (inlet water temperature) on the performance of a CO_2 heat pump has been investigated by Yokoyamaa et al. (2007). At minimal inlet water temperatures of 5–15 °C, the maximal achieved water temperature was 60 °C at a pressure of 102.8 bar in the gas cooler. In the research done by Fernandez et al. (2010), the operation conditions for the heat pump include a temperature of evaporation of 10 °C, and a maximal pressure within the gas cooler of 110 bar. Zhang et al. (2010) also reported experimental studies on the optimum pressure within a heat exchanger for a CO_2 heat-pump system. The minimal temperature within the evaporator is 10 °C and the maximal pressure within the gas cooler is 125 bar. The isentropic efficiency of the compressor is in the range of 0.65–0.7.