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Development and validation of a finite element model for water – CO_2 coaxial gas-coolers

D. Sánchez [a], R. Cabello [a,*], R. Llopis [a], E. Torrella [b]

[a] Department of Mechanical Engineering and Construction, Campus de Riu Sec., Jaume I University, E-12071 Castellón, Spain
[b] Department of Applied Thermodynamics, Camino de Vera 14, Polytechnic University of Valencia, E-46022 Valencia, Spain

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ABSTRACT

The gas-cooler heat exchanger is a characteristic device of the refrigeration plants that use CO_2 as the working fluid in transcritical conditions. The proper gas-cooler sizing, allows getting the maximum effectiveness in the heat exchange, thus permitting to reduce the temperature approach and improving the heat recovered in the coolant (secondary) fluid and the COP of this kind of systems.

This work deals with the development of a steady-state coaxial gas-cooler model that uses water as coolant fluid. The model is based on the finite-volume methodology and is validated with experimental data. An uncertainty lower than ±10% was obtained in the calculation of the heat transfer rates and an uncertainty lower than ±3 °C in the outlet temperatures of both fluids. Several correlations for estimating the CO_2 heat convection coefficient have been studied to select the one that best fits the model. Using the model, the influence of different parameters on the gas-cooler thermal effectiveness has been analyzed and presented in this work. Finally, a comparison between the traditional ε-NTU method and the developed model has been done.

1. Introduction

The growing concern about the contribution of the HFC fluids to the greenhouse effect has encouraged the seeking of environmental friendly refrigerants. Natural refrigerants such as propane (R290), isobutene (R600a), ammonia (R717) or carbon dioxide (R744) constitute the most considered alternatives. The studies carried out by some authors dealing with the advantages and disadvantages of these fluids [2,3], have placed CO_2 as a clear alternative to the HFCs, mainly because of the combination of its thermophysical/security properties and its low cost [4,5]. Despite its advantages, in some specific applications, the energy efficiency achieved with CO_2 transcritical cycles is lower than that of conventional HFCs cycles. That introduces a handicap that is being overcome by improving the behavior of different components or introducing new configurations [6,7].

One of the most important components in CO_2 transcritical cycles, is the heat exchanger named gas-cooler, which cools the refrigerant in transcritical conditions with a high thermal effectiveness [8] reducing the necessary heat transfer area [9]. Since the energy efficiency of the refrigeration system mainly depends on the enthalpy of the refrigerant at the evaporator inlet, the thermal effectiveness (or in other words, the temperature approach: $T_{GC,o} - T_{W,i}$) is a key parameter to improve this efficiency. Accordingly, the development of accurate gas-cooler models is essential to achieve a better dimensioning to improve the heat transfer process and the COP of the transcritical cycles [10].

In the open literature, there are various gas-cooler models for heat pumps and refrigeration cycles using air [9,11–14] or water [14,15] as a secondary fluid. In all of them, the gas-cooler model is used to simulate or to improve the complete behavior of a refrigerating plant or heat pump, however only a few of them analyze the gas-cooler model development and its validation [9,13]. Accordingly, this work is devoted to complete the analysis of gas-cooler models developing and validating a steady-state mathematical model of coaxial gas-cooler working with R744 and water as secondary fluid in counter flow arrangement. The model is based on the finite-volume methodology in order to take into account the important variations of the thermophysical properties near the pseudocritical region [1]. Different correlations for estimating the CO_2 convective coefficient have been evaluated with the model. The results calculated using the model have been compared with the experimental ones obtained after 351 tests.

Furthermore, and using the model, the influence of several parameters on the gas cooler thermal effectiveness has been analyzed: water and refrigerant inlet temperature, water and refrigerant mass flow rate, and gas-cooler inlet pressure.

* Corresponding author. Tel.: +34 964 728135; fax: +34 964 728106.
E-mail address: cabello@emc.uji.es (R. Cabello).

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Nomenclature

A	heat transfer surface (m^2)
A_C	water area (m^2)
c_P	isobaric specific heat capacity ($kJ\ kg^{-1}\ K^{-1}$)
d, D	diameter (m)
d_{eq}	equivalent diameter ($d_{eq} = d_o \cdot Nt^{0.5}$) (m)
D_H	hydraulic diameter (m) ($D_H = 4 \cdot A_C \cdot P_W^{-1}$) (m)
e	relative error (%)
h	enthalpy ($kJ\ kg^{-1}$)
k	thermal conductivity ($W\ m^{-1}\ K^{-1}$)
L	volume finite length (m)
$\dot{m}$	mass flow rate ($kg\ s^{-1}$)
Nt	number of tubes
NTU	number of transfer units
Nu	Nusselt number (–)
P	pressure (MPa)
Pr	Prandtl number (–) ($Pr = c_P \cdot \mu \cdot k^{-1}$)
P_W	wet perimeter (m)
$\dot{Q}$	heat transfer rate (kW)
Re	Reynolds number (–) ($Re = \rho \cdot D_H \cdot v \cdot \mu^{-1}$)
s	standard deviation
T	temperature (K)
V	speed ($m\ s^{-1}$)
$\dot{v}$	volumetric flow rate ($m^3\ h^{-1}$)

Greek symbols

α	convection heat transfer coefficient ($W/m^2\ K$)
ρ	density ($kg\ m^{-3}$)
ε	thermal effectiveness (%)
μ	dynamic viscosity (Pa s)

Subscripts

Actual	actual
Avg	average value
b	bulk conditions
Dis	discharge
BP	back-pressure
GC	gas-cooler
Glic	water-ethylenglycol mixture (50% in vol.)
i	inlet/inner
j	iterator
Max	maximum
Met	metal
Mot	electrical motor
n	number of finite volume
o	outlet/outer
O	evaporator
Opt	optimum
ps	pseudocritical conditions [1]
R	refrigerant
Rec	liquid receiver
Suc	suction
W	water

2. Experimental plant

The experimental refrigeration plant used to carry out the necessary tests to validate the model is shown in Fig. 1.

The system is equipped with a semihermetic reciprocating compressor (displacement: 3.48 m^3/h at 1450 rpm), a two stage expansion system with a refrigerant vessel between stages [16], and two coaxial heat exchangers (evaporator and gas-cooler) with counter flow arrangement. In both heat exchangers, the main fluid is the refrigerant R744 and the secondary fluids are water in the gas-cooler and a water-ethylenglycol mixture (50% in vol.) in the evaporator.

The different plant operation conditions are achieved through two auxiliary systems, one that supplies the thermal duty to the evaporator and another that removes the thermal load from the gas-cooler. Both systems are provided with flow and outlet temperature controls [17].

The plant monitoring is carried out with T-type thermocouples and pressure gauges placed at the inlet and outlet of each element. All temperatures are measured on the pipe surface, except for the compressor discharge temperature which is measured with an immersion thermocouple. A Coriolis-effect mass flow meter measures the refrigerant mass flow rate at the gas-cooler outlet, while two electromagnetic flow meters measure the volumetric flow rates of secondary fluids at the evaporator and gas-cooler outlets. The compressor power consumption is measured with a digital wattmeter, while its speed is obtained from the inverter drive that governs the compressor. Table 1 shows the calibration range and the uncertainties of each sensor and measuring device.

The geometrical dimensions of the gas-cooler are shown in Fig. 2. The heat exchanger bundle is made up of seven smooth copper tubes (a) enclosed in a carbon steel tube (b). The CO_2, due to its high pressure, flows through the inner tubes, while water flows through the outer tube in counter current. This flow arrangement is adopted for security reasons and is proper for energy recovery applications. To avoid the heat losses to the environment the external surface of the gas-cooler has been isolated (c), so it can be assumed that all the heat removed from the refrigerant is transferred to the water.

3. Development of the finite-volume gas-cooler model and experimental validation

3.1. Methodology

Thermophysical properties of CO_2 in the pseudocritical region suffer sudden variations [1]. This characteristic represents an important difficulty to construct the model, since it is not possible to consider a constant value of those properties along the heat exchange process. To avoid this drawback, the heat exchanger has been modelled using the finite-volume methodology, considering steady-state operation and constant thermophysical properties in each finite-volume corresponding with the average ones inside each finite-volume [9].

Carbon dioxide and water properties (density, enthalpy, etc) are obtained with the Refprop dynamic libraries [18] linked to Matlab©, which is the platform used for programming the model.

Fig. 3 shows the data-flow diagram of the model, the four modules that make it up and their interconnections.

From the input data ($T_{W,i}$, $T_{GC,i}$, $\dot{m}_W$, $\dot{m}_R$, P_{GC} and the dimensions of the heat exchanger), the model is initialized using modules 1 and 2. In this process, the inlet and outlet variables of each finite-volume are established for the first time. For this, the water temperature in each volume is considered to be equal to the temperature of the water at the inlet of the gas-cooler ($T_{W,i}$) and the refrigerant temperature is considered to be equal to half of the