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Two-phase loop thermosyphon using carbon dioxide applied to the cold end of a Stirling cooler

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HIGHLIGHTS

- Two-phase loop thermosyphon using CO_2 was experimentally studied for Stirling cycle.
- A maximum in heat conductance was achieved when superheat was close to zero.
- Maximum heat conductance obtained was 46.72 W K^{-1}.
- 85% of the results from a first principle model developed fell within ±20% band.

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ABSTRACT

This work focuses on an experimental investigation of a carbon dioxide thermosyphon loop designed to fulfill the geometric and temperature requirements of a specific FPSC (Free Piston Stirling Cooler). Experiments were carried out varying the temperature difference between the heat source, i.e. air at the entrance of the evaporator and heat sink, i.e. internal surface of the condenser, refrigerant charge and evaporator airflow rate. The experimental results were explored using the thermal conductance concept applied to each heat exchanger and also to the whole loop. It was found that the loop was able to carry a maximum heat transfer rate of 514 W with a heat source-sink temperature difference of 11 °C. A first principles model was also developed and 85% of the calculations fell within ±20% of measurements. Further exploration of the model showed a good capability to capture trends and be an aid in the thermosyphon design phase.

1. Introduction

Due to the shortage of the electric energy sources and also to the strict environmental controls regarding the use of HFC and HCFC refrigerants, non-conventional cooling technologies — thermoelectric, magnetocaloric, thermoacoustic, Stirling cycle, etc. — are now being considered as serious alternatives, most of them with innovative features regarding the thermodynamic efficiency and the use of environmental friendly substances. However, all those new ways of producing cold are indirect systems, meaning that they generate a cold and a warm end in the refrigeration machine. This creates a need for secondary loops to carry heat between the cold end and the refrigerated compartment and also between the warm end and the outdoor air.

Recently, Hermes and Barbosa, [7]; reported a methodology for a thermodynamic comparison of thermoelectric, Stirling and vapor compression cycles for portable coolers. The comparison was carried over using own generated experimental results at ambient temperatures of 21 °C and 32 °C and also data from Ref. [6]. The Stirling cooler studied had a thermosyphon on the cold end and an air source forced convection finned surface at the warm end. They concluded that the Stirling and the vapor compression systems had similar thermodynamic efficiencies of about 14%, while a linear vapor compression system had about 8% and the thermoelectric cooler was at 1%. This proves the thermodynamic competitiveness of the Stirling cooler with the vapor compression technology.

To avoid a degradation of the system coefficient of performance (COP), any secondary heat transfer loop should operate with a minimum heat source-sink temperature difference and also with a

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Nomenclature

A	area (m^2)
C	thermal capacity ($W\ K^{-1}$)
D	internal diameter (m)
f	friction factor (−)
G	gass flux ($kg\ m^{-2}\ s^{-1}$)
g	gravity ($m^2\ s^{-1}$)
h	heat transfer coefficient ($Wm^{-2}\ K^{-1}$)
k	thermal conductivity ($Wm^{-1}\ K^{-1}$)
L	length (m)
m	mass flow rate ($kg\ s^{-1}$)
N	number of elements in discretization (−)
NTU	number of transfer units (−)
Nu	Nusselt number (−)
P	pressure (kPa)
Pr	Prandtl number
Q	heat transfer rate (W)
R	thermal resistance (WK^{-1})
Re	Reynolds number (−)
T	temperature (°C)
UA	thermal conductance (WK^{-1})
x	quality (−)
z	height (m)

Greek symbols

α	correction coefficient for heat loss (−)
β	correction constant for heat loss (−)
Δ	difference (−)
ε	effectiveness (−)
η	Fin/Finned surface efficiency (−)
μ	viscosity ($kg\ m^{-1}\ s^{-1}$)
ρ	density ($kg\ m^{-3}$)

Subscripts

air	air stream
ch	cold head
cond	condenser
conv	convection
eq	equivalent
evap	evaporator
ext	external
h	hydraulic
H	high temperature
$H–C$	high−low temperature difference
in	inlet
int	internal
INLET	inlet to test section in wind tunnel
l	liquid
m	microchannel tube
max	maximum
min	minimum
OUTLET	outlet to test section in wind tunnel
r	ratio
ref	refrigerant
surf	surface
total	total
tubes	microchannel tubes
v	vapor
wall	condenser inner wall

minimum pumping power. Examples of such loops are: loop heat pipes (LHP), capillary pumped loops (CPL) and two phase loop thermosyphons (2PLT).

Some works addressing the development of secondary heat transfer loops for Stirling coolers may be found in the open literature [4,10], most of them applied to low cooling capacity refrigeration systems (less than 100 W). In this particular work, Berchowitz and collaborators have developed a 40 W cooling capacity unit, utilizing a thermosyphon with carbon dioxide as means for the heat transfer in the cold head of the FPSC. Results reported show better performance than thermoelectric and vapor compression systems. Larsson et al. [11] reported the results on the application of an FPSC to a domestic freezer using thermosyphons for both sides. Propane was used on the warm side and carbon dioxide was used on the cold side. They were able to achieve −11 °C inside the cabinet while the carbon dioxide thermosyphon operated at −18 °C and the cold head at −37 °C. Oguz and Ozkadi, [14] applied an FPSC to a domestic refrigerator having an R134a thermosyphon on the cold head and an extended surface with forced ventilation on the warm side of the FPSC. They were able to achieve temperatures ranging from 4.3 °C to 8.5 °C and their calculated COP was stated to be from 1.37 to 1.89, however a closer inspection on their results yields to corrected actual COP ranging from 0.71 to 0.99. Patent activity includes a specific case by Rudick and Berchowitz, [18] where a refrigeration system to cool a refrigerated cabinet is claimed to have a Stirling cooler and a thermosyphon connected together.

In the sense of developing the understanding of two phase loop thermosyphons, the works of McDonald et al. (1977) Ali and McDonald, [2], and [12] differentiate themselves for its broadness and completeness. They carried out experiments and developed a computational tool to simulate the thermal behavior of a 2PLT. Effects of refrigerant charge, tilt angle, heat source-sink temperature difference, tubing length and inner diameter where all explored in their work. Considerable high values of thermal conductance over a wide range of temperature differences were found, by a proper choice of the tubing diameter and length and refrigerant charge. It was also shown that the operational limit of the system is determined by the dry-out of the evaporator.

Considering the use of carbon dioxide in two-phase loop thermosyphons and closed thermosyphons, an increased interest has been found lately not only in refrigeration but also on power generation and heat pumping. Rieberer, [17] lays out an 8 year summary of its experience in developing natural circulating probes and collectors for ground source heat pumps. In this work the carbon dioxide probes are closed thermosyphons with 15 mm o.d. and each probe has a heat transfer rate of about 0.9 kW, adding up to about 7.5 kW in total capacity. The heat collectors using carbon dioxide were tested in a laboratory environment and they are similar in concept to the two-phase loop thermosyphon of interest to the present work. It was shown a strong dependence of the heat transfer rate and the temperature difference in the collector head (CO_2 condenser) on the charging amount, which translates into the area of the evaporator that experiences two-phase flow. Jeong and Lee, [9] report on an experimental analysis of a two-phase carbon dioxide filled thermosyphon aiming at geothermal heat recovery. They used a 9.9 mm i.d. x 1 m long thermosyphon in their experiments. Temperature differences of up to 7 °C and heat transfer rates of up to 125 W were observed.

Earlier work was also conducted at the University of Illinois in Prof. Predrag S. Hrnjak's group by Gustavo Weber, in 2002, using a parallel flow microchannel evaporator and a microchannel annular condenser. Even though in private communication the results are