2010

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ABSTRACT

A free-piston Stirling cooler was implemented into a 27 liter mini refrigerator. Two different fluids were employed as the working fluid. The tested fluids were water and a nanofluid which consisted of dispersed aluminum oxide particles with a volume fraction of 23%. The circulation of the working fluids in the loop was driven by a submersible pump. A jet-impingement heat exchanger was designed, built and tested with three different fluid types for comparison. The unique heat exchanger with two inlets (one circular jet and one rectangular slot) and a single outlet was implemented on the cold side of the Stirling cooler. On the experimental side, temperature readings at the critical locations of the cooling system were recorded using T-type thermocouples and a data acquisition system. For the sake of repeatability, both fluids were tested for five times and the temperature values obtained from each location were averaged among the five tests. In addition, the flow cases for both fluids were simulated using a commercial CFD software. A close agreement in between the experimental and numerical analyses was observed.

1. INTRODUCTION

In the past few decades, the negative effects of conventional vapor compression refrigeration systems on the nature has been noticed and therefore studies on Stirling coolers, which are alternative systems, have grown up. Free piston Stirling coolers operate with helium, an inert gas, as their working fluid. The helium gas inside the cooler is compressed back and forth by the coupled movement of the piston and the displacer. As a result; the heat load absorbed from a lower temperature ambient, is rejected to a higher temperature ambient. Since the Stirling coolers can have variable cooling capacities due to input voltage, they may be distinguished as similar devices to the variable capacity compressors. Another significant advantage of the Stirling coolers is that they can keep their high efficiencies even in low cooling capacities.

Basically; closed loop of working fluid and/or fan systems may be used in the application of free piston Stirling coolers to refrigeration units. In closed loop systems, a working fluid of choice (i.e. water, nanofluid, etc.) is circulated through the system by aid of a circulation pump. The loop is constructed by connecting the heat exchanger inside the refrigerator to the heat exchanger that is attached to the cold head of the cooler, allowing oneway flow to the working fluid within the system. In fan systems, extended surface heat exchangers are used and ambient air is blown or sucked over the heat exchangers by a fan.

In Stirling cooler applications, the most critical component is the cold side heat exchanger which requires a highly efficient design due to space limitations. Hence, significant amount of research is being focused on this part. A Stirling cycle cooled domestic refrigerator was demonstrated by Oguz and Ozkadı (2002). In this application; a thermosyphon heat exchanger with an annular condenser was used on the cold side of the cooler, whereas a forced convection mechanism was built up on the warm side, using an extended surface heat exchanger and a fan. 5°C average cabinet temperature was maintained while the power consumption of only the Stirling cooler was 30.5 W and the energy consumption was 732 Wh/24h at 25°C ambient temperature. Celik (2003) achieved the same cabinet temperature with a daily energy consumption of 636 Wh/24h, using a microchannel heat exchanger on the cold head of the cooler.

Welty and Cueva (2001) reported a study on the application of two thermosyphon systems to a domestic freezer. According to the test results; -18°C average cabinet temperature was achieved where the outside ambient temperature was 25°C. 40 % decrease in energy consumption has been calculated, when compared to the conventional vapor compression systems. Another application with two thermosyphons has been carried out by Green *et al*. (1996) from Oxford University. An 80 liter cabinet was used in the tests as a freezer and -20°C average cabinet temperature has been achieved with a power consumption of 1480 Wh/24h. When compared to equivalent conventional vapor compression systems, energy saving of 17 % has been yielded.

Jet-impingement heat exchangers have found wide applications especially in cooling of hot surfaces for effective removal of locally concentrated heat. Chupp *et al*. (1969), Metzger *et al*. (1972), and Tabakoff and Cleveneger (1972) studied the measurement of impinging cooling on a semi-circular surface with an array of round jet. They also examined the effects of distance between the jet and the target surface. Hrycak (1981) found that the heat transfer at the stagnation point on a concave surface is higher than that for a flat plate, due to increased surface area. In this study, a convex surface at a colder temperature than the fluid was employed where the fluid is being cooled and the target surface is being heated. A jet-impingement heat exchanger with two inlets (a circular jet and a rectangular slot) was designed for improved efficiency.

2. EXPERIMENTAL SETUP

A free piston Stirling cooler was integrated to a 27 liter mini refrigerator. The prototype was built up by using; a Stirling cooler, a refrigerator cabinet, a circulation pump, a jet-impingement heat exchanger system on the cold side and a forced convection mechanism on the hot side of the Stirling cooler. The cooler and its cold head are shown in Figure 1.

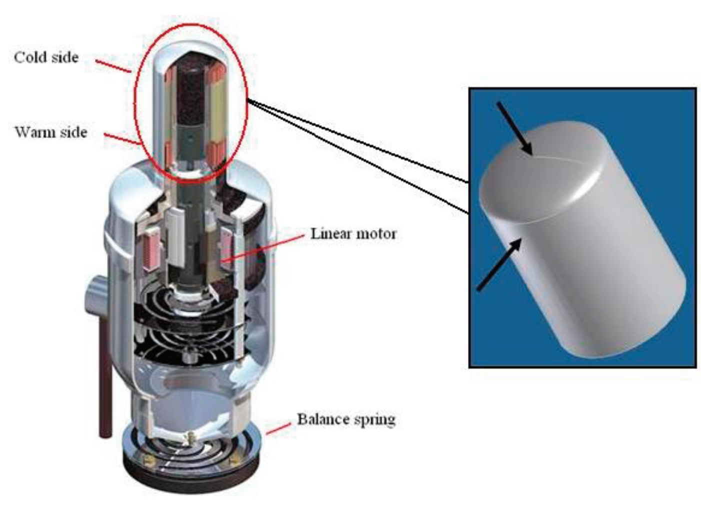

Figure 1: Stirling cooler and its cold head

The unique jet-impingement heat exchanger designed in this study has two inlets and an outlet. There is a rectangular slot (7mm x 2mm) inlet on the side and a circular jet (d = 3 mm) on the top of the heat exchanger both spraying the fluid towards the cold head of the cooler as given in Figure 1 above. The schematic of the heat exchanger is illustrated in Figure 2.

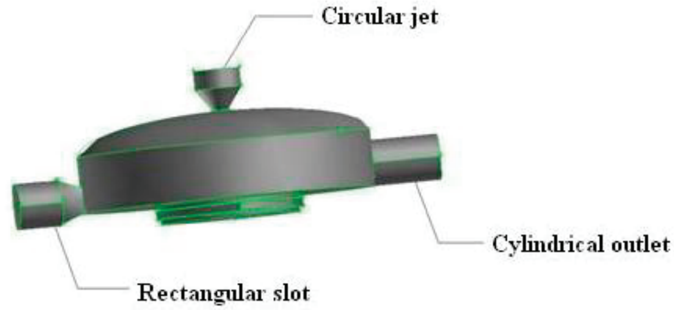

Figure 2: Jet-impingement heat exchanger

Two different working fluids were tested and compared. One of the working fluids was water, while the other one was a nanofluid having aluminum oxide particles with a mean volume particle size of 0.16 µm. The particles were mixed with water and had a volume fraction of 23% in water. Mixing of the nano particles and water was conducted by the manufacturer by colloidal dispersion with the dispersant.

For the temperature measurements, nine T-type thermocouples were used at critical locations of the experimental setup, while one was used for recording the ambient temperature of the test room. The temperature values were acquired by a data acquisition system. The measurements were ensured to be within ±0.25°C. For all of the tests, the sampling period for the temperature measurements was five seconds. Figure 3 shows the schematic of the experimental setup.

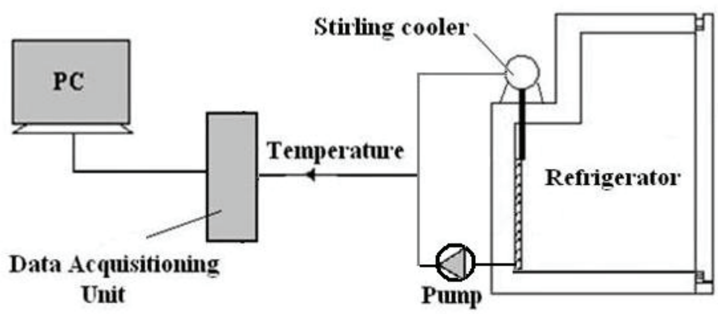

Figure 3: Schematic of the experimental setup

The Stirling cooler which operates based on a variable capacity linear motor (0-12V), was set at 12V for maximum cooling capacity in both sets of experiments with water and the Al_2O_3 nanofluid. The tests were to be continued for approximately three hours and the steady-state periods were examined in all of the tests.

3. THEORETICAL ANALYSIS

For the working fluid flows of interest, the flow fields were characterized using the conservation laws. In this analysis, both fluids were assumed to be incompressible and to behave as a Newtonian fluid with constant properties. The flow was considered to be a 3-D flow with constant heat flux from the bottom surface of the heat exchanger where the cold head of the Stirling cooler is placed. Hence the continuity, momentum and energy equations of the flow were simplified, respectively, to:

Continuity:

$$\frac{1}{r}\frac{\partial(ru)}{\partial r}+\frac{1}{r}\frac{\partial(v)}{\partial\theta}+\frac{\partial(w)}{\partial z}=0 \tag{1}$$

r-momentum:

$$\rho\left(\frac{\partial(u)}{\partial t}+u\frac{\partial(u)}{\partial r}+v\frac{\partial(v)}{\partial \theta}+w\frac{\partial(w)}{\partial z}-\frac{v2}{r}\right)= -\frac{\partial(p)}{\partial r}+\mu[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial(u)}{\partial r}\right)+\frac{1}{r^2}\frac{\partial^2 u}{\partial \theta^2}+\frac{\partial^2 u}{\partial z^2}-\frac{u}{r^2}-\frac{2}{r^2}\frac{\partial(v)}{\partial \theta}+\rho g \quad (2)$$

θ-momentum:

$$\rho\left(\frac{\partial(v)}{\partial t}+u\frac{\partial(v)}{\partial r}+\frac{v}{r}\frac{\partial(v)}{\partial \theta}+w\frac{\partial(w)}{\partial z}+\frac{uv}{r}\right)= -\frac{1}{r}\frac{\partial(p)}{\partial r}+\mu[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial(v)}{\partial r}\right)+\frac{1}{r^2}\frac{\partial^2 v}{\partial \theta^2}+\frac{\partial^2 v}{\partial z^2}+\frac{2}{r^2}\frac{\partial(u)}{\partial \theta}-\frac{v}{r^2}+\rho g] \quad (3)$$

z-momentum:

$$\rho\left(\frac{\partial(w)}{\partial t}+u\frac{\partial(w)}{\partial r}+\frac{v}{r}\frac{\partial(w)}{\partial \theta}+w\frac{\partial(w)}{\partial z}\right)= -\frac{\partial(p)}{\partial z}+\mu[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial(w)}{\partial r}\right)+\frac{1}{r^2}\frac{\partial^2 w}{\partial \theta^2}+\frac{\partial^2 w}{\partial z^2}+\rho g] \quad (4)$$

Energy:

$$\rho\frac{\partial}{\partial t}(E)+\nabla.\left(v(\rho E+p)\right)= \nabla.\left(k_{eff}\nabla T-\textstyle\sum_j h_j J_j+(\tau_{eff}.v)\right) \quad (5)$$

For comparing the thermal performances of the working fluids, coefficient of performance (COP) of both systems can be calculated by:

$$COP=\frac{Q_{cap}}{W_{net}} \quad (6)$$

where Q_{cap} is the cooling capacity of the system which under steady state conditions is equal to Q_{leak} which is the rate of heat leaking into the system. Q_{leak} can be obtained by:

$$Q_{leak}=UA_{cabinet}(T_\infty-T_{cabinet})=Q_{cap} \quad (7)$$

where $UA_{cabinet}$ is the overall heat transfer coefficient for the refrigerator and T_∞ and $T_{cabinet}$ are the ambient and average cabinet temperatures, respectively. The UA value for the refrigerator was obtained experimentally as 1 W/K by using the reverse heat leak method.

4. RESULTS

Temperature readings from the critical locations for both experiment sets are given in Figure 4. Number of the data on the graph is reduced for the sake of the clarity of markers.

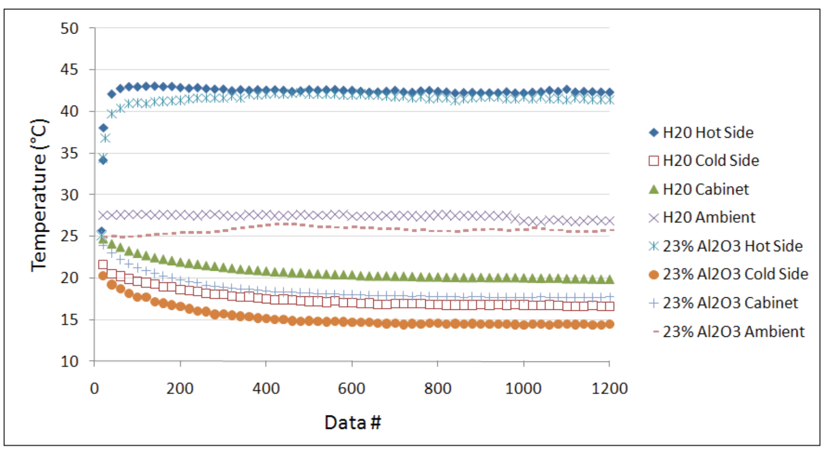

Figure 4: Temperature data from the cooler, cabinet, and ambient

Working fluid temperatures at the main inlet and outlet of the jet-impingement heat exchanger were also recorded for comparison. The temperature behaviors of both fluids throughout the experiments are illustrated in Figure 5.

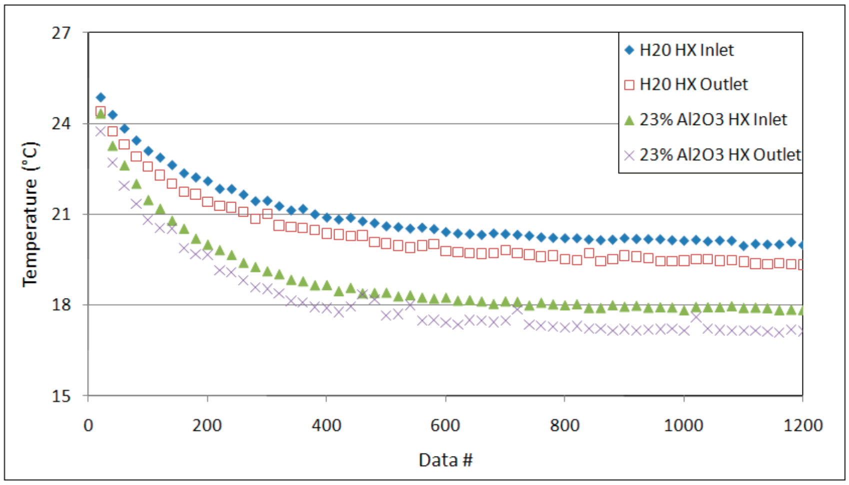

Figure 5: Temperature data from the heat exchanger

Temperature and velocity distributions within the jet-impingement heat exchanger were obtained using Fluent software. The results were found to be in close agreement with the experimental results. Temperature and velocity behaviors of fluid particles at the early stage of the fluid circulation are given in Figures 6 and 7. These sketches represent the values at the mid-plane of the jet-impingement heat exchanger.

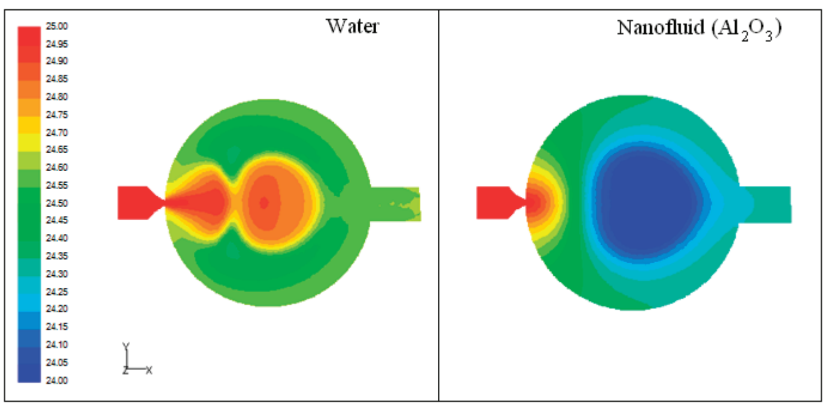

Figure 6: Temperature distribution (°C) at t=15s at the mid-plane of the heat exchanger

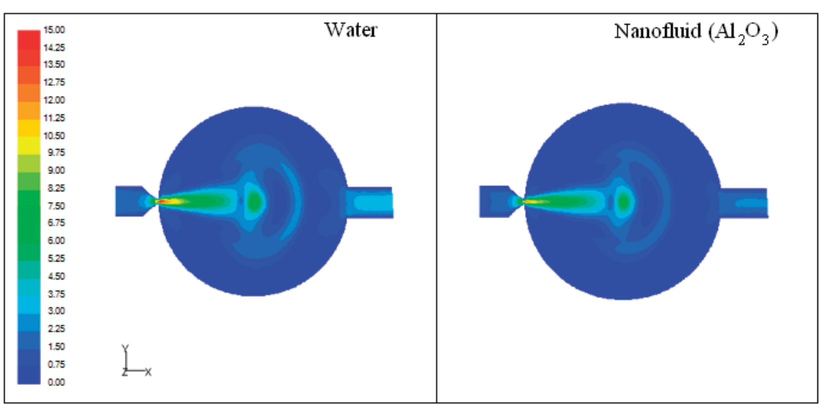

Figure 7: Velocity distribution (m/s) at t=15s at the mid-plane of the heat exchanger

The Stirling cooler with the fan on its warm side had a total power of 40W, and the circulation pump had an input of 5 W, resulting in total input power of 45 W. The cooling capacities with the water and the nanofluid tests were calculated as described in the previous section and were found to be 7 W and 8 W, respectively. COP values of both systems were calculated, and are illustrated in Figure 8.

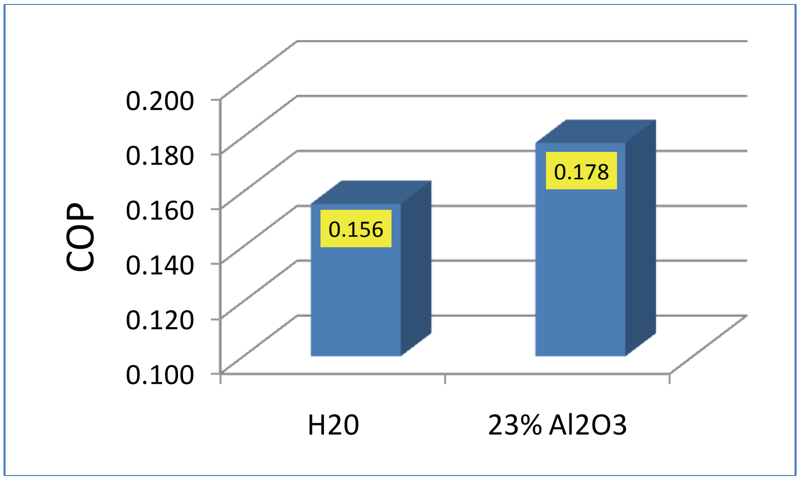

Figure 8: COP comparison

5. CONCLUSIONS

A free piston Stirling cooler with a jet-impingement heat exchanger on its cold head was implemented into a refrigerator of 27 liter volume. Two different working fluids, water and a nanofluid (23% Al_2O_3 in H_2O), were tested and the results were compared both experimentally and numerically. Experimental results showed that temperature change of the nanofluid through the heat exchanger was 16.6% higher than that of water. This agreed with the CFD simulation results. Under same conditions with the Stirling cooler at maximum capacity (40W), and the circulation pump with 5W of power input, 7W and 8W of cooling capacities were achieved with the water and nanofluid tests, respectively. Cooling capacities were calculated by assuming that they are equal to heat leak into the system per conservation of energy law. Overall heat transfer coefficient value for heat leak calculations was determined experimentally by reverse heat leak method. The overall COP of the system employing nanofluid was found to be 14.1% higher than the performance of the system using water.

NOMENCLATURE

COP	coefficient of performance	(-)
E	energy	(J)
h	specific enthalpy	(J/kg)
k	thermal conductivity	(W/m.°C)
p	pressure	(kg/m.s^2)
Q	heat transfer rate	(W)
r	radius	(m)
t	time	(s)
T	temperature	(°C)
UA	overall heat transfer coefficient	(W/°C)
V	velocity	(m/s)
W	power	(W)
μ	viscosity	(kg/m.s)
ρ	density	(kg/m^3)
τ	shear stress	(kg/m.s^2)

Subscripts

cap	capacity
eff	effective
leak	heat leak
net	net
∞	ambient

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ACKNOWLEDGEMENT

The authors would like to thank Mr. Brenton Baker, Mr. Patrick Keller, Mr. Matthew Bilbruck, Mr. Zachary Stead, and Mr. Kristopher Long, who designed and built the heat exchangers, and helped in building the experimental setup for this study.