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The design of ARCTIC: A rotary compressor thermally insulated μcooler

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Abstract

Microscale cooling to date relies largely on passive on-chip cooling in order to move heat from hot spots to alternate sites. Such passive cooling devices include capillary pump loops (CPL), heat pipes, and thermosiphons. Recent developments for active cooling systems include thermal electric coolers (TECs) for heat removal. This paper focuses on the design of an active microscale closed loop cooling system that utilizes the Rankine vapor compression cycle. In this design, a rotary compressor will generate the high pressure required for efficient cooling and will circulate the working fluid to move heat away from chip level hot spots to the ambient. The rotary compressor will leverage technology gained from the rotary engine power system (REPS) program at UC Berkeley, most specifically the 367 mm^3 displacement platform. The advantage of a Wankel (Maillard) compressor is that it provides six compression strokes per revolution rather than a single compression stroke common to other popular compressors such as the rolling piston. The current Wankel compressor design will achieve a theoretical compression ratio of 4.7:1. The ARCTIC (a rotary compressor thermally insulated μcooler) system will be a microscale hybrid system consisting of some microfabricated (or MEMS) components including microchannels, in plane MEMS valves, and MEMS temperature, pressure and flow sensors integrated with mesoscale, traditionally machined steel components, including the compressor itself. The system is designed to remove between 45 W of heat at 1000 rpm using R-134a but the system is easily scaleable through a speed increase or decrease of the compressor. Further, a vapor compression cycle using R-134a operating between 258 and 310 K has a theoretical coefficient of performance (C.O.P.) of approximately 4.6. While this calculation does not include pressure losses, compressor inefficiency, or heat transfer losses, it provides ample room for significant improvement over comparable TECs with C.O.P.s of approximately 0.1–0.2. Finally, a thermal circuit analysis determines that the time constant to achieve refrigeration temperature at the evaporator in 12 s is possible.

Keywords: MEMS; Compressor; Refrigeration; Electronic cooling; Vapor compression; Heat transfer; Rakine cycle

1. Introduction

The vapor compression cycle is the most common mechanical-type refrigeration cycle. The cycle and its variants are found anywhere from the common household refrigerator to a vehicle's air conditioning system. However, on the microscale, the cycle has not been employed as more often than not passive designs such as thermosiphons, heat pipes [1], and capillary pump loops (CPL) [2], have been developed. Active systems, such as thermoelectric coolers (TEC) [3], have been used as alternatives as well. These other options have found their way into the microscale in lieu of the vapor compression cycle for two reasons. The first reason applies only to the passive designs: the vapor compression cycle requires input power. Passive designs generally rely on a fluid's phase change ability to remove heat and to also provide motion. However, these designs are temperature limiters in reality and are limited to the fluid's thermal properties in regards to the final temperature of the cooled chip.

The second reason vapor compression cycles have been limited on the microscale is the size of the system. Each of the previous designs, active or passive, have been minimized to some degree of success. While several pumps have been put to use on the microscale [4], they have been limited to the amount of pressure head which can be added to the fluid. Further, compressors have been limited on the microscale mostly due to entropy considerations [5].

However, success at UC Berkeley in the rotary engine power system (REPS) program has shown that engines can be operated at small scales [6]. This program has specifically demonstrated a 367 mm^3 displacement engine is capable of combustion. The

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Nomenclature

A	cross-sectional area heat travels through (m^2)
c_p	specific heat for constant pressure (J/kg K)
C	thermal capacitance (J/K)
C.O.P.	coefficent of performance
h	heat transfer coefficient (W/m^2 K)
i	electrical current (A)
k	thermal conductivity (W/m K)
K	constant
L	conducting length, length heat will travel (m)
m	mass (kg)
P	pressure (N/m^2, atm)
Q	heat (W)
r_p	pressure ratio
r_v	compression ratio
R	Thermal resistance (W/K)
t	time (s)
T	temperature (K)
V	volume (m^3)
$\mathcal{V}$	voltage (V)

Greek symbols

γ	ratio of specific heats
ε	emissivity
σ	Stefan–Boltzman constant (W/m^2 K^4)

Subscripts

cond	thermal conduction
conv	thermal convection
in	into the compressor
out	out of the compressor
rad	thermal radiation
s	surface of device
sur	surrounding air

cycle of this rotary engine is intake, compress, ignite, expand, and exhaust. A compressor only requires an intake, compression, and exhaust stroke. Therefore, the development of a rotary compressor driven design is ongoing [7].

Rotary compressor sealing has long been troublesome at the small scale for high compression applications [8]. Sealing at the small scale is difficult because the differential pressure across the seal is larger and as the scale is reduced the dimensions are small, making the pressure gradient large. Small scale refrigeration systems on the other hand, have two primary advantages: (1) the requisite pressure ratio is reduced and (2) the compression stage in vapor compression systems occurs near the liquid boundary and any liquid present enhances the sealing by reducing the effective leakage area. The wet compression process can offer some thermodynamic advantages, however in this particular example, it is not explicitly considered.

The initial compressor design has a footprint of 25 mm × 30 mm or about the size of one Intel® Pentium® 4 chip. To enable a vapor compression cycle with an ideal coefficient of performance (C.O.P.) around 4.6 a moderate pressure ratio of 4.7:1 is required, where C.O.P. is defined as

$$\text{C.O.P.} = \frac{\text{heat lift}}{\text{power in}}$$

In this analysis, the processes are assumed to be isentropic and the performance of real systems will be lower. This isentropic approach was carried out rather than using polytropic compression because of a lack of experimental data. As such a sensitivity analysis has been carried out. Despite the reduction in performance due to leakage and entropy generation the theoretical C.O.P. is significantly greater than thermoelectric coolers.

Thermoelectric coolers have seen rapid development advances with the introduction of enhanced nanostructures [9]. The current figure of merit (ZT) for one thermoelectric material, Bi_2Te_3/Sb_2Te_3, is 2.4, which is a significant increase. The figure of merit sets a maximum temperature across the TEC element. This maximum temperature drop is however, at the point of no heat pumping (i.e. $Q = 0$). The attained temperature drop decreases with increased thermal load (Q), and therefore, TECs are typically used in stacked modules to move significant quantities of heat. Furthermore, TEC's suffer from parasitic conduction of heat back to the cool side of the device because the materials used are inherently conductive, both electrically and thermally, limiting their C.O.P. Vapor compression cycles can minimize this by using insulating layers to limit parasitic conduction losses while maintaining a high C.O.P.

This paper will discuss the design of a complete vapor compression refrigeration cycle for the microscale which can theoretically achieve heat lifts of 45 W at 1000 rpm at a temperature change of 40 K from the ambient. This will include the design of individual parts as well as a thermal circuit analysis using lump capacitance modeling. The thermal circuit analysis will be developed not only to show the performance of the system but also analyze parameters which will lead to optimal performance of the system.

2. ARCTIC design

The vapor compression cycle is composed of four essential steps. The active part of the system, the compressor, receives uncompressed refrigerant and does work on this vapor by compressing the gases to the desired pressure. The compressor releases the vapor into the passive part of the system, beginning with the condenser. In the condenser, the refrigerant goes through a phase change, transforming from a vapor to a liquid. In the process, the refrigerant releases heat to the environment. Once the refrigerant leaves the condenser it is in a high pressure, high temperature liquid state prepared to enter the expansion valve. The expansion valve expands the liquid into a low pressure, low temperature state. The refrigerant moves through the evaporator absorbing heat by again changing phases, this time from liquid to vapor. Then the cycle repeats [10]. A review of this cycle can be seen in Fig. 1. Work is ongoing to optimize the compressor design, however, the details of this work will not be discussed in detail here.