High-performance, low-cost liquid micro-channel cooler

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ABSTRACT: This paper presents test results for a high-performance, low-cost micro-channel cooler, which is made by cold-working copper strip. No metal removal results from the fin formation, so no copper scrap is produced by the fin forming process, as occurs in machining or photo-chemical etching processes. The micro-channel cooler has 0.54 mm fin pitch and 2.1 mm fin height and was tested with water in laminar flow. Tests were performed for 1-pass and 2-pass flow configurations. Also tested was a commercially available macro-channel cooler (Thermaltake Bigwater 735) which operates in the turbulent flow regime. Performance comparisons are provided for the two tested coolers. The micro-channel design gives higher thermal performance at 1/6 the flow rate of the macro-channel cooler and has ¼ the fluid pumping power. The micro-channel cooler is much smaller and lighter in weight than the commercial the macro-channel cooler.

1 INTRODUCTION

Liquid cooling for desktop computers has arrived! The main commercial applications today are for gaming computers. There have been many publications on use of liquid cooling for CPU cooling. Many of these publications have focused on water cooled micro-channels. The micro-channels conceivably may be made on the back side of the silicon chip, or copper micro channels can be used as an attached external metal sink. An example of micro-channel cooling analysis is provided by Upadhye and Kandlikar [4].

This paper describes a new technology micro-channel cooler for cooling computer and server central processor units (CPU). The present methodology of making the micro-channels provides a low cost cooling device that operates at very high thermal performance. A typical micro-channel consists of very small parallel channels (e.g., rectangular cross-section) formed in a copper plate. The desired channel width is on the order of 50-500 microns. Current technology to make such small channels may use photo-chemical etching or other micro-machining techniques. Although such channels could conceivably be machined, milling methods cannot easily be used to make such small channels. The time required to mill such small channels would be so long that the resulting micro-channel cooler would be very costly. Electro-static discharge machining (EDM) is another possible approach, although this is also very costly. Both milling and EDM methods result in the generation of scrap material resulting from the metal removed to form the channels. A metal embossing operation is one possibility for forming channels in the 100-500 micron width range. However, a limitation of this method is that the desired channel height cannot be formed. For best performance, one desires a channel height 5-to-10 times the channel width. Further, one desires the fin thickness that separates the channels to be quite small – on the order of 100-250 microns.

Examination of the present commercial applications for desktop liquid cooling shows that liquid coolers typically does not use "micro-channels." Rather, they use "macro-channels" machined into a copper block. An example is the Thermaltake "Bigwater 735" waterblock [Thermaltake, (2006)], which is shown in Figure 1. The 4.5 mm wide x 10 mm deep serpentine channel is milled from a copper block. This device operates in the turbulent flow regime. The overall dimensions of the Figure 1 waterblock are 77×70 mm, plus a 48×5 mm round spreader on the base. The overall thickness of the copper block is 13.5 mm. Intel (2006) has also developed a liquid cooling device. Both the Thermaltake and Intel liquid cooling solutions operate at a substantially higher liquid flow rate than that of the present micro-channel cooler. The Intel cooler operates at 3.78 l/min and the measured balance point of the Thermaltake cooler is 2.28 l/min.

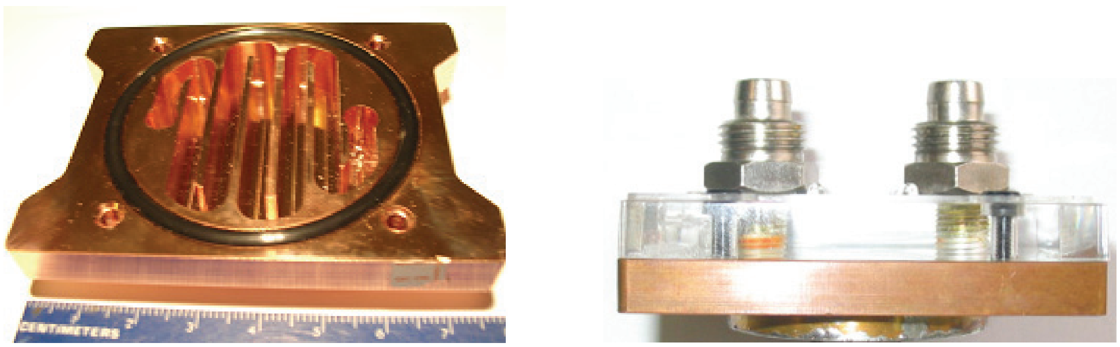


Figure 1 a) Thermaltake "Bigwater" waterblock having one serpentine channel 4.5 mm wide x 10 mm deep, b) Side view with Plexiglas cover and fluid connectors.

Figure 2. Copper micro-channels used in present work, described as "Fin-H."

2 PRESENT MICRO-CHANNEL CONCEPT

The channels are formed by a high-speed proprietary operation without generation of any scrap material. A key feature of the present technology is to make the fins thin and high. The channel height divided by the channel width is defined as the "channel aspect ratio." The fin height divided by the fin thickness is defined as the "fin aspect ratio." It is highly desirable that both the channel aspect ratio and the fin aspect ratio be large. A high channel aspect ratio provides large surface area for heat transfer, while a large fin aspect ratio will minimize material content. The coolant pressure drop is also influenced by the channel aspect ratio. A large channel aspect ratio is desired to minimize the coolant pressure drop. Because the channels are so small, they will operate in the laminar flow regime for practical applications. The channels used in the present work are shown in Figure 2 and are designated as geometry “Fin-H.” These channels have 7.4 channel aspect ratio and 8.0 fin aspect ratio. The channel hydraulic diameter (D_h) is 0.491 mm.

A second factor in selection of the micro-channel cooler design is the plan area and the liquid flow circuiting. Combined analytical and experimental evaluation of the Figure 2 micro-channels resulted in selection of a micro-channel 25 mm wide with 20 mm flow length. This is sufficient to provide high performance heat removal for the current generation of desktop computers. If needed, the flow length could be increased to obtain higher performance.

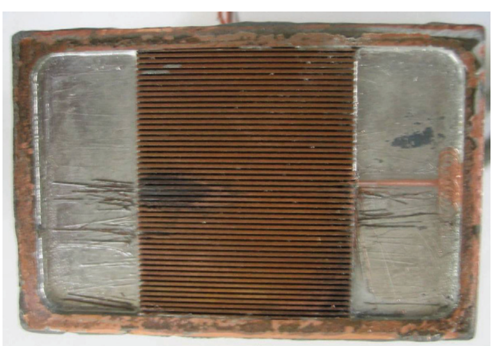
Both single-pass and two-passes on the water side were evaluated analytically and experimentally. Because the flow is in the laminar regime, the coolant heat transfer coefficient is independent of coolant velocity. However, a 2-pass design will have about 4-times the pressure drop of a 1-pass design. A key concern of the 1-pass design is whether flow mal-distribution will exist in the multiple parallel channels. Experimental results on 1-pass a 2-pass designs showed no loss of thermal performance in the 1-pass design, relative to the 2-pass design. Hence, the 1-pass design is preferred, because of its much lower coolant pressure drop.

Figure 3 shows a prototype configuration of the 1-pass design having 20 mm micro-channel flow length. This sample was made with 10 mm flow length in the entrance and exit manifolds. However additional work showed that the preferred version has 7.0 mm flow length in the entrance and exit manifolds.

3 EXPERIMENTAL PROGRAM

Figure 3b shows the test configuration for the micro-channel cooler. This consists of the Figure 3a micro-channel cooler, a 45 mm long, 16 mm square cross-section copper block, and a copper heater block, in which two 150 W, 6.35 mm diameter, electric cartridge heaters are inserted. Three 1.0 mm diameter Type K thermocouples are inserted in holes drilled at 5.0 mm pitch in the 16 mm square copper block. The thermal conductivity of the copper block is known and is 391 W/m-K. The heat flux is measured by the temperature gradient in the copper block, and hot surface temperature at the micro-channel surface is obtained from extrapolation of the temperature gradient measured by the three thermocouples. The tests were done using water flow in the micro-channels.

The material thickness below the Figure 3b micro-channels is 2.5 mm, and is 1.5 mm in the manifold water inlet and exit regions.



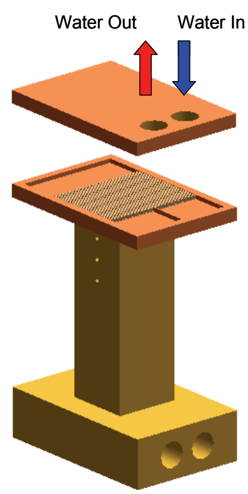


Figure 3. a) Prototype test sample of 1-pass micro-channel cooler with 20 mm micro-channel flow length, b) Test setup illustrated for a 2-pass design.

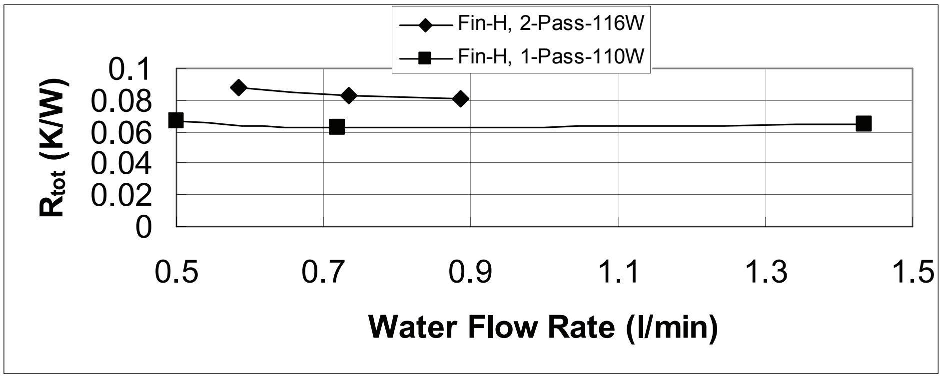


Figure 4. R_{tot} of 1-pass vs. 2-pass micro-channel coolers.

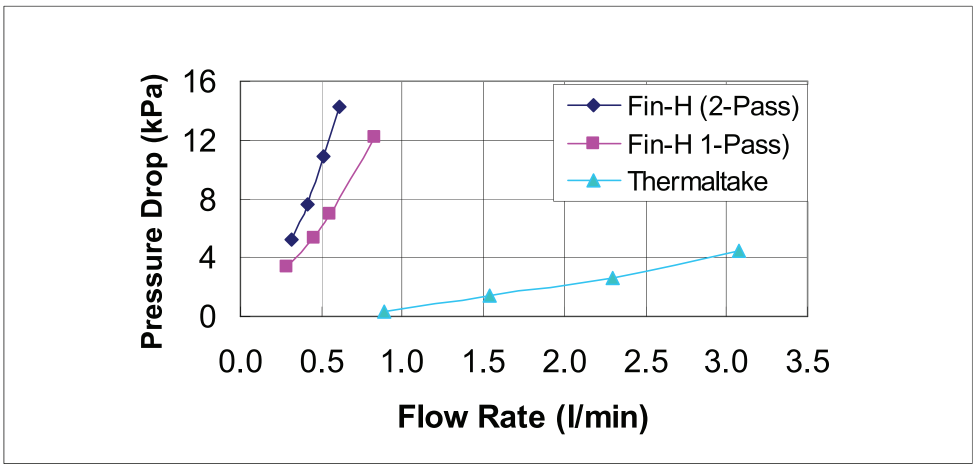


Figure 5. Measured pressure drop for the 1-pass and 2-pass micro-channel design and for the Figure 1 Thermaltake "waterblock."

Figure 4 shows the measured thermal performance of the micro-channel cooler (Fin-H) for approximately 110 W input. The R_{tot} is defined as

$$R_{tot} = \Delta T_{lm} / q \tag{1}$$

where ΔT_{lm} is the log-mean temperature difference, which is defined in terms of ($T_{hot} - T_{c,out}$) and ($T_{hot} - T_{c,in}$). Note that R_{tot} is NOT defined in terms of the ITD, as frequently found in the electronic cooling literature. Use of the conventional Equation (1) definition results in R_{tot} = 1/UA. Reasons for defining R_{tot} in terms of ΔT_{lm} are discussed by Webb (2007).

The R_{tot} is composed of two component thermal resistances, the micro-channel convection resistance (R_{conv}) and the spreading resistance (R_{sp}). The thermal conduction resistance in the 2.1 mm high fins is neglected, because the calculated fin efficiency was approximately 98%.

Calculation of the spreading resistance is ambiguous, because the spreader thickness is 2.5 mm in the 20×25 mm micro-channel region and 1.5 mm in the manifold region. However, the R_{sp} and R_{conv} were iteratively calculated using the experimental R_{tot} 16×16 mm source area, and 20×25 mm spreader area. The equation of Lee et al. (1995) was used to calculate the spreading resistance. This calculation showed that R_{conv}/R_{tot} = 0.52.

Figure 4 shows that R_{tot} = 0.063 K/W for the 1-pass design and 0.08 K/W for the 2-pass design. The author was initially concerned that the 1-pass design may yield higher R_{tot} than for the 2-pass design, because of possible flow mal-distribution. However, as shown in Figure 4, this is problem did not occur. The flow rate was well within the laminar flow regime for all test points. The highest Reynolds number was 522, which was for the highest flow rate with the 2-pass design.

Figure 5 shows the measured pressure drop for the 1-pass and 2-pass designs. The pressure drop of the 1-pass design is only 38% that of the 2-pass design. Because the 1-pass design gives lower R_{tot} and lower pressure drop, this design is preferred. Figure 5 also shows the measured pressure drop for the Figure 1 Thermaltake Bigwater 735 cooler. The measured balance point in the Thermaltake cooling system was found to be at 2.28 l/min.

4 DISCUSSION

Figure 6 shows the measured R_{tot} for the present 1-pass micro-channel cooler and the Thermaltake cooler. At the 2.28 l/min system balance point, the Thermaltake cooler provides R_{tot} = 0.106 K/W. The operating point selected for the present 1-pass micro-channels is 0.36 l/min – only 16% of the flow rate in the Thermaltake waterblock.

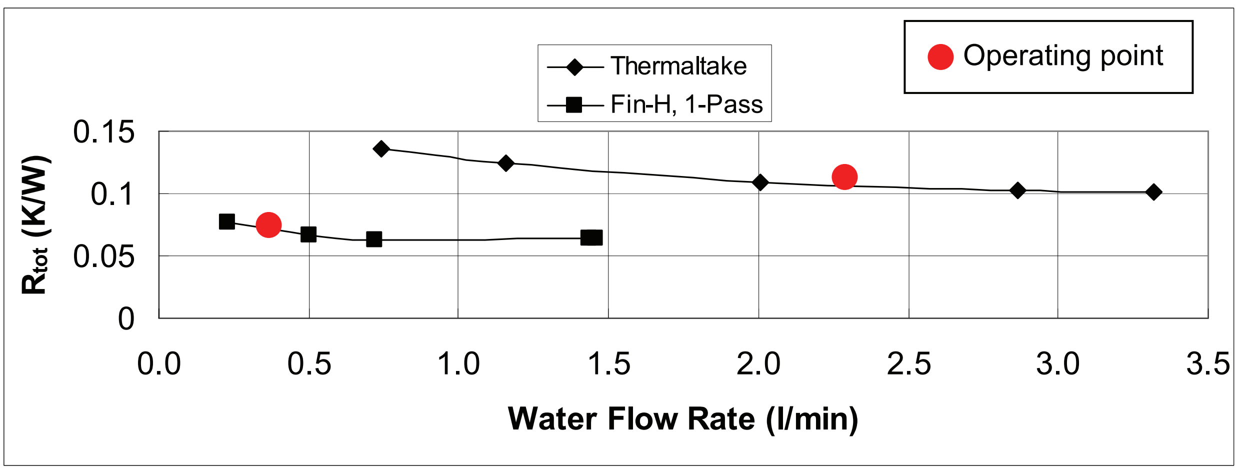


Figure 6. Comparison of R_{tot} for 1-pass micro-channel cooler and Thermaltake cooler.

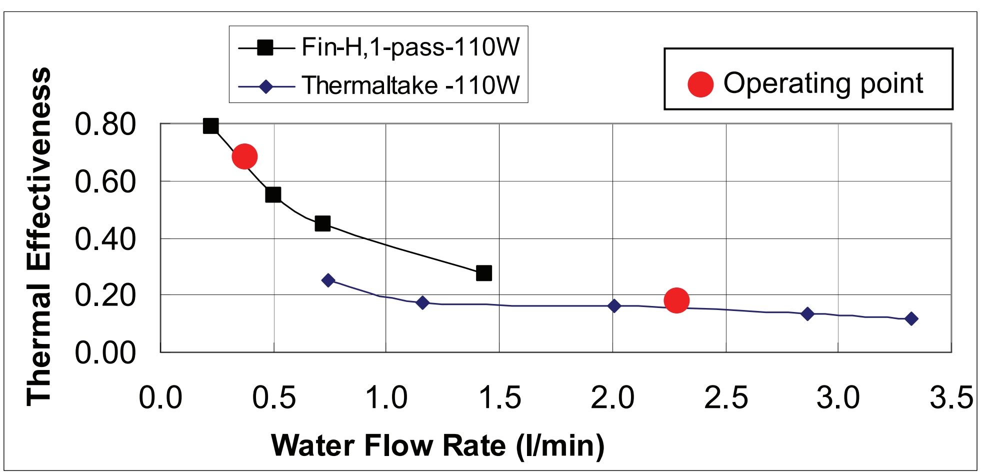


Figure 7. Thermal effectiveness vs. water flow rate for micro-channel cooler vs. Thermaltake.

Figure 7 shows heat-exchanger thermal effectiveness (ε) of the present 1-pass micro-channels and the Thermaltake cooler. Note that the Thermaltake operates at very low thermal effectiveness (15%) at its 2.28 l/min balance point. At the 0.36 l/min operating point selected for the present 1-passmicro-channel design, the thermal effectiveness is 0.63.

Table 1 compares the present 1-pass micro-channel thermal performance with that of the Thermaltake cooler at the operating point of each design. At the respective operating points, the present micro-channel design gives 29% lower R_{tot} and 60% higher water pressure drop. However, the pumping power ratio (volume flow rate × pressure drop) of the present micro-channel design is only 25% that of the Thermaltake cooler.

Table 1. Comparison of performance at 110 W input.

Geometry	Flow (l/m)	R_{tot} (K/W)	Thermal Effect.	Δp (kPa)	Flow Power N-m/s^2
Fin-H (1-pass)	0.36	0.075	0.65	4.20	0.025
Thermaltake	2.28	0.105	0.15	2.62	0.100

The micro-channel cooler product design is illustrated in Figure 8. Comparison with the Figure 1Thermaltake cooler shows that the Figure 8 design is much smaller and lighter in weight. The compact Figure 8 design may also be applied for cooling of CPUs in servers.

Although it is possible to make an all aluminum design similar to Figure 8, the approximately 50% lower thermal conductivity of aluminum design will not have the performance of the all copper design. Further, it would be difficult to attach aluminum micro-channels to a copper heat spreader without an added interface thermal resistance. In addition, water can be used as the working fluid in a copper cooler, without a corrosion inhibiter, as would be required in an aluminum cooler.

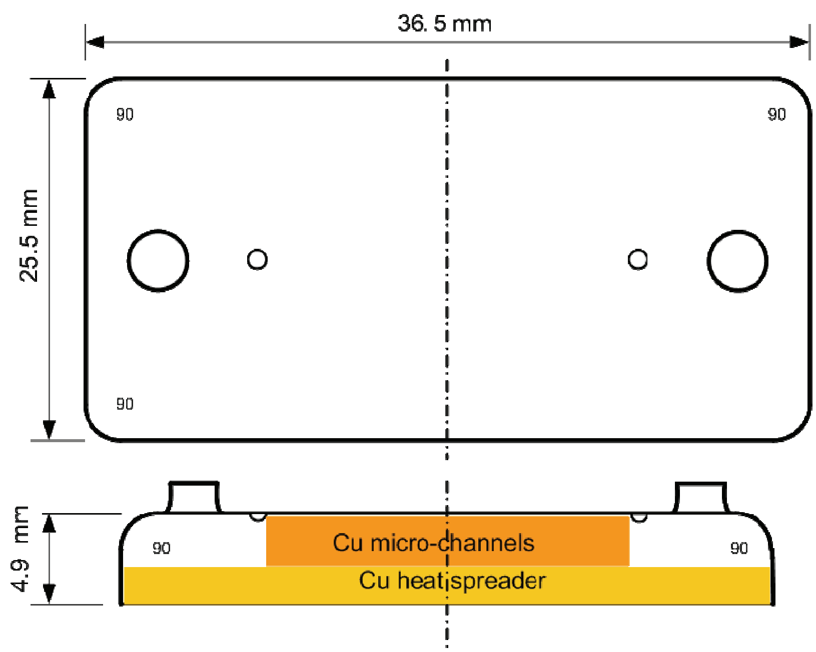


Figure 8. Illustration of fabricated micro-channel cooler.

5 CONCLUSIONS

This paper describes a compact, high performance all copper micro-channel cooler that is smaller, and lower weight than the Figure 1 "macro-channel" cooler. The copper micro-channel cooler provides lower thermal resistance and operates at about 1/6 the coolant flow rate and 25% of the fluid pumping power as the Figure 1 macro-channel cooler.

The all-copper micro-channel cooler is applicable to cooling electronic heat sources in desktop computers, servers, and other applications (e.g., military electronics). The Figure 2 micro-channel plate may be formed with a different flow length or number of channels.

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NOMENCLATURE

A	Heat transfer convection surface area
D_h	Hydraulic diameter
h	Heat transfer coefficient
ITD	Inlet Temperature Difference (T_{hot} - $T_{c,in}$)
$\dot{m}$	Mass flow rate
q	Heat transfer rate
R_{conv}	Convection thermal resistance (= $1/\eta hA$)
R_{sp}	Spreading thermal resistance.
R_{tot}	$\Delta T_{lm}/q$.
$R_{tot,ITD}$	$(T_{hot} - T_{c,in})/q$.
Re	Reynolds number, Re_{Dh} (based on hydraulic diameter)
T_c	Coolant temperature. $T_{c,in}$ (inlet coolant), $T_{c,out}$ (outlet coolant)
T_{hot}	Hot source temperature
U	Overall thermal resistance.
Δp	Coolant pressure drop.
ΔT_{lm}	Log-mean temperature difference.
ε	Heat exchanger thermal effectiveness [$=T_{c,out} - T_{c,in})$/ITD]
η	Surface efficiency for finned surface

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