

An Experimental Study on Enhancing Cooling Rates of Low Thermal Conductivity Fluids Using Liquid Metals

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Abstract: In a previous numerical study (Al Omari, Int. Communication in Heat and Mass Transfer, 2011) the heat transfer enhancement between two immiscible liquids with clear disparity in thermal conductivity such as water and a liquid metal (attained by co-flowing them in a direct contact manner alongside each other in mini channel) was demonstrated. The present work includes preliminary experimental results that support those numerical findings. Two immiscible liquids (hot water and liquid gallium) are allowed experimentally to exchange heat (under no-flow conditions) in a stationary metallic cup where they are put in direct contact. The experimental results confirm the significant heat exchange enhancement. The superior thermal conductivity of the used liquid metal as compared with the water is the reason behind the observed enhancement in heat transfer. For the same residence time and the same contact surface area between the two liquids, however, the experiments show a slightly slower rate in the heat transfer between the two liquids compared to the case of the channel flow considered in the numerical simulations. This discrepancy is justified on the basis of the additional enhancements brought about by the forced convective effects in the case of the channel flow (which are absent in the experiments where conduction and natural convection prevail).

Keywords: Enhanced heat transfer, liquid gallium, water, interfacial dynamics, immiscible liquids, mini-channel flow, high thermal conductivity liquids.

1 Introduction

The augmentation of heat exchange is of clear importance in many industrial applications, specifically the cooling of electronics. The major motivation behind our previous numerical and the current experimental research is to explore and validate

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methods that may find acceptance and future applications in areas that might be requiring intensive heat transfer removal, particularly in electronics cooling industry. In a previous work (Al Omari S.-A. B.,2011) a numerical study was conducted on the enhancement of heat transfer in small scale channels between liquids with clear disparity in their thermal conductivity (water and mercury). In the present work experiments were conducted to shed further light on the prevailing heat transfer phenomena that take place at the interface between such liquids.

There are different approaches suggested by many other researchers to perform effective thermal management and control of heat generated in electronics systems (Bar-Cohen A., Kraus A. D., Davidson S. F.,1983; Incropera F. P.,1988; Yeh L. T, 1995; Sathe S., Sammakia B, 1998; Chomdee S., Kiatsiriroat T., 2006; Tsai T.-E., Wu H.-H., Chang C.-C., Chen S.-L.,2010; Wits W. W., Vaneker T.H.J., Mannak J.H., Legtenberg R, 2009). It is anticipated that the approach and results presented in this paper, in addition to the previous published work (Al Omari S.-A. B., Haik Y., Abu Jdayil B.,2010; Al Omari S.-A. B., 2011) can contribute to such systems, among others. Meanwhile, although our current and previous works were motivated with electronics cooling being the main target in mind, the techniques and approaches adopted in these researches can be applied to other applications, wherever effective exchange of heat is required. Al Omari's results (Al Omari S.-A. B., 2011) show that heat transfer in small-scale channels through which hot fluids with relatively moderate heat conduction capabilities such as water, can be enhanced effectively by co-flowing these lower conductivity liquids with other fluids that have superior heat conduction capabilities; e.g. liquid metals.

For the sake of making ease of reference and to facilitate comparison between the present preliminary experimental findings with our earlier numerical studies, some brief numerical results are also presented in this paper. In the numerical part discussed, liquid gallium is used as a heat transfer agent to enhance heat transfer from hotter water in a direct-contact co-flow system in a small scale channel. Physical and thermal properties for gallium compared to other liquids, e.g. water and mercury, are given in table 1 (Ma K.-Q, and Liu J,2011).

Table 1: Thermo physical Properties of used fluids

Thermal Property.	Water	Gallium	Mercury
The density "kg/m ³ "	1000	6096	13470
Specific heat capacity "J / (kg K)"	4183	381.5	138.2
Thermal conductivity "W / (m K)"	0.6	28.7	8.4
Dynamic Viscosity "kg / (m s)"	0.00101	0.00181	0.0014

The basis for selecting water in this study as the fluid to be cooled with the help of superior heat-conducting agents (gallium in this case) is supported by fact that water is increasingly becoming a common fluid in electronic cooling applications (Dumont G., Fontaine Vive Roux Ph., Righini B., 2000; Incropera F. P., 1999; Naphon P., Wiriyasart S, 2009; Nguyen C. T., Roy G., Gauthier C., Galanis N, 2007; Kwang-Soo, K., Myong-Hee, W., Jong-Wook, K., and Byung-Joon, B, 2003). In this sense, water is allowed to circulate in the heat generating components and subsequently this heat is to be removed from the water in such a manner that the water can be reused again in a closed circuit cooling system. The experimental set up used in the present study is discussed below.

2 The Experimental Part

The experiments were carried out by pouring an assigned amount of hot water into a small metallic cup containing a known amount of liquid gallium, at room temperature. The bottom of the cup is attached to a large water bath which is used as a heat sink, the side walls of the cup were directly exposed to the surrounding room air (See Fig. 1). The cup is open from the top and so the hot water in the cup will also be exposed to direct contact with the surrounding air from its upper side. The majority of the heat that is lost by the water, however, is transferred to the gallium through the interface between the two liquids. Subsequently, the heat will penetrate through the gallium body and eventually pass to the outer water bath and the surroundings. A smaller portion of the heat will be lost through the upper part of the cup wall which is in direct contact with surrounding air.

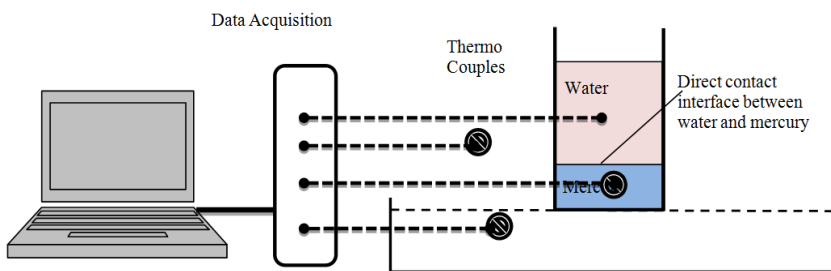


Figure 1: Schematic Drawing of the Experimental Setup

Four measurements were conducted to report the temperature history of the liquid gallium, water in the cup, water bath, and the surrounding air. The thermocouples used to measure the gallium and water temperature were located at different rele-

vant points in each of the fluid bodies. The temperature of the water bath heat sink was measured by a thermocouple located near the bottom of the metallic cup. The temperature measurements were collected by using the LABVIEW data acquisition program with a sampling rate of one measurement per second. The locations of the thermocouples in the two liquids were selected in such a way that representative average temperatures for both liquids would be obtained. The history of cooling of the hot water and at the same time the heating up of the gallium with time is recorded. At the end, the amount of heat lost from the water is calculated based on the temperature measurements and the application of energy balance.

3 The Numerical Analysis Part

Only a brief description of the numerical problem addressed will be given here as more details can be found in our previous work (Al Omari S.-A. B.,2011). Some results based on the numerical simulations will be compared with the experimental findings and used to help analyze the experimental results obtained in the present study aiming at attaining further insight into the heat transfer processes considered. Additional numerical results will be presented to aid a more comprehensive understanding of the processes at hand.

In Reference (Al Omari S.-A. B.,2011) hot water and a liquid metal (mercury) exchange heat in a mini-channel; see Fig. 2, the two liquids flow alongside concurrently in the channel in a direct contact manner. The liquid metal receives some of the heat from the water and it subsequently passes it through the channel wall to the outer surroundings. The wall material of the channel is assumed to be one of a high thermal conductivity such that the assumption of constant uniform wall temperature might be justified. Implicit here is probably the need for a fan to cool down the channel wall from the outer side that is in direct contact with the surrounding air, so that channel wall will stay at a temperature close enough to the surrounding air temperature without any significant temperature non-uniformity or buildup in the wall body.

3.1 *Mathematical and physical formulation*

The methodology of the numerical part is presented and discussed in our previous work (Al Omari S.-A. B.,2011) . Hence, this methodology will only be described here briefly. Pressure-velocity coupling is accomplished by the SIMPLE algorithm. The flow and interaction between the two liquids used, which are representing two immiscible phases, are described by the volume of fluid (VOF) model. Details about the VOF model can be found in the widely open literature (Kothe D. B., Rider W. J., 1995; Tang H, Wrobel L. C., Fan, Z, 2004). The VOF model gives volume fraction values for each fluid in each computational cell. The volume fraction for

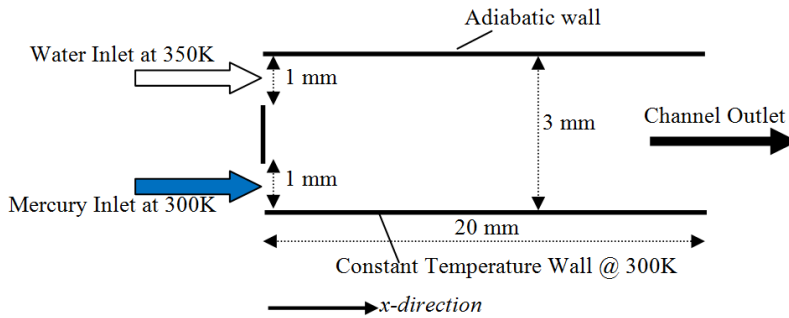


Figure 2: Schematic of the Channel System used in the numerical simulations

each fluid ranges from 0 to 1. If in any computational cell the value for volume fraction was calculated to be between these limits then this indicates that in that cell there is an interface between the two fluids. Overall values for the thermo-physical properties of the fluid mixture in each cell are decided based on the volume fraction of each fluid in the cell. The CFD code Fluent 6.3 (see The CFD package Fluent 6.3) has been utilized to solve the momentum, energy, and the VOF model equations.

The interface between the two fluids is tracked based on the values of the volume fraction of each of the two phases that can be deduced by solving the continuity equation for the volume fraction of one of the liquids and then obtaining the volume fraction of the other by applying the constraint that the sum of volume fraction must be unity at any location in the flow domain. For details on the mathematical formulation, see (Al Omari S.-A. B.,2011) . Convergence of the numerical solution is judged based on the normalized residual for the solved governing equations. For the continuity equation a value of 1×10^{-5} is considered to be the limit. Lower residual values for the other equations were attained by the time when the continuity equation has converged. For further details on the numerical and mathematical formulation, see (Al Omari S.-A. B.,2011) . Mesh independency analysis of the numerical runs reported is presented in the subsequent results section.

4 Results and Discussion

The experimental results are presented and discussed in this section. The conclusions drawn from the numerical study part and their agreement with the conceptual findings of the experiments will be highlighted. Table 2 summarizes the conditions of the conducted experiments in which liquid gallium is used to aid cooling down

hotter water while being in direct contact with it during the cooling process in a stationary metallic cup, under no flow conditions (see Fig. 1).

Table 2: Conditions of the experimental tests

Test No.	Water amount (ml)	Volume of liquid gallium used (ml)
1	40	0
2	40	40
3	40	60

Figure 3 shows the time variation of the measured temperature of the water in the cup, observed experimentally in the three tests considered. The figure clearly shows a faster cooling rate of the water in the presence of liquid gallium in tests 2 and 3 as compared with water cooling rate in test 1, in which water is cooling down while being alone in the cup unaided by the liquid metal. Both tests 2 and 3 show almost identical results (both qualitatively and quantitatively). This indicates that the actual amount of gallium present in the cup plays no big role regarding heat transfer rate, and that what matters most is the real being and existence of the gallium in the experiments, setting in a direct contact manner with the water. More testing to further quantify of the effect of the actual amount of gallium used on the overall heat transfer rates, might still be needed before final conclusions can be drawn in this regard. However, for the time being this is not within the scope of the present paper and is under consideration in an on-going project by the present authors.

A more detailed quantitative analysis of the differences between the different tests, are presented in Tables 3 and 4 which compare between the heat transferred from the water in test 1 (with no aid by the gallium) with the heat lost from the water in the other tests aided with gallium, at different times during the experiments. Once more, the results of both Tests 2 and 3 are almost identical and the basic test with no gallium (Test 1) shows a clear shortcoming in terms of the rate of heat transfer from the water, especially during the initial stages of the cooling process. During the first 100 seconds of the test time, gallium results in rates of heat transfer from the water higher by at least 50 percent than the heat transfer in the baseline case with no gallium (i.e. Test 1). At later times, due to the reduced temperature difference between the water and the surroundings, the heat transfer becomes slower in all cases and the superiority of the tests with gallium over the test without gallium reduces (in the average) to about 30 percent in the subsequent 50 seconds (see Table 4). By that time (i.e. after 150 seconds from the onset of the tests) water has already lost about 65 percent of its initial thermal energy content, while by that

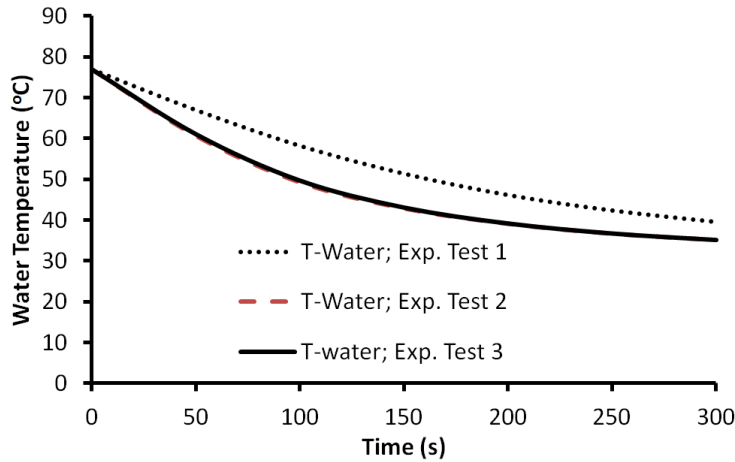


Figure 3: Variation of the water temperature with time in the three experimental runs

time the loss in Test 1 is around 50 percent. The percentage of superiority of the tests with gallium continues to decline with time such that after around 500 seconds from the onset of the test (not shown in the results) all tests show almost the same rate of heat loss and at that time both liquids reach almost thermal equilibrium with the surroundings (i.e. completely lost their thermal potential relative the surroundings). In an attempt to relate the above to real-life cooling systems for which the concepts addressed in this paper might be relevant, the above discussion highlights the importance of the residence time allowed for the hot fluid (hot water in this case) to lose heat, and hence will be reflected on the compactness of the final heat transfer channel system built. In this regard, it can be concluded that the use of a liquid metal such as gallium based on the concepts highlighted in this and in our previous works (i.e. by direct contact with the less conducting fluids), can lead to the development of clearly smaller and more effective heat removal systems, without the need to circulate these liquid metals necessarily themselves in the channels of the cooling system developed. This will allow for using more feasible liquids (e.g. water); that have acceptable user, environment, and economical features, in a better way than already existing practices, by allowing them to dump the heat they removed from the to-be-cooled parts to a sink, much more effectively.

In what follows we refer to the numerical simulation results of the flow of water and gallium in the mini channel considered in Al Omari S.-A. B.(2011) (see Fig. 2) and later on we will try to relate these results to the findings of the experiments

Table 3: Percentage of heat loss from water in the different experimental tests

Time (sec)	Percentage of heat lost during 150 seconds		
	Test 1 Water 40ml, <i>No liquid metal</i>	Test 2 Water 40ml, Gallium 40ml	Test 3 Water 40ml, Gallium 60ml
30	10.8%	18.0%	18.2%
60	22.0%	35.9%	36.4%
90	33.2%	48.8%	49.2%
120	42.2%	58.3%	58.5%
150	49.7%	65%	65.2%

Table 4: Heat loss in the tests using liquid gallium as compared to the baseline test with only water

Time (sec)	Heat loss in Test 2/Heat loss in Test 1	Heat loss in Test 3/Heat loss in test 1
30	1.667	1.685
60	1.632	1.655
90	1.470	1.482
120	1.382	1.386
150	1.310	1.332

conducted in the present work. Table 5 summarizes the details and the conditions of the two numerical tests considered (Tests 1A and 1B). The two numerical tests in Table 5 are classified based on the ratio of the mass flow rate of gallium to the mass flow rate of the water concurrently flowing in the channel.

For both numerical cases, the inlet temperatures of the water and the liquid metal were kept at 350K and 300K, respectively. These temperatures, more or less, are close to the starting hot water temperature and the environment temperature in the experiments. The bottom wall of the channel, which is in contact with the liquid metal, is taken to be at a fixed uniform temperature of 300K; assumed to be similar to the temperature of the surroundings. The details of the flow and heat transfer for Case 1A are compared with those corresponding to case 1B. These two cases are used here to demonstrate the salient features of the flow and heat transfer phenomena taking place in the channel, with and without gallium.

Table 5: Conditions of the numerical tests;

Case No	R= Mass ratio of liquid metal/water	Inlet speed of water (cm/s)	Liquid metal used
1A	0; Only water	0.68	None
1B	2	0.68	Gallium

Some details of the flow field are presented in Fig. 4, where the velocity profiles at a downstream station a distance 10 mm from the channel inlet, are presented. Fig. 4 shows almost a parabolic profile for Case 1A, irrespective of the small recirculation region established in the corner of the backward step at the channel inlet (not shown in the results; for details see Al Omari S.-A. B. (2011)). In case 1B, slight deviation from the parabolic profile at the interface between the two liquids is observed due to the interface between the two immiscible liquids at hand.

Fig. 5 shows the temperature profile for the two cases 1A and 1B, at 10 mm downstream from the inlet. In case 1A, the temperature varies in the expected manner, where it increases gradually from the lowest temperature value at the wall to the highest temperature value at the other side where the adiabatic wall boundary condition is implemented. The temperature gradient starts at maximum level near the colder wall side and gradually declines until it reaches negligible levels near the opposite adiabatic wall. The temperature gradient, which is the driving force for heat transfer from the flowing liquids and the cold wall, is presented in Fig. 6. The variation of the temperature gradient for case 1A is in accordance with the above description.

The temperature profile for case 1B is presented in Fig. 5. Due to the very high thermal conductivity of liquid gallium compared with water, the temperature distribution in the thin liquid gallium body next to the cold wall is very uniform and is very close to the value set at the colder boundary, around 300 K. The temperature varies sharply in the interface region between the two liquids, which indicates high temperature gradient, and hence high rates of heat transfer in that region between the two liquids. Due to the high temperature gradient in case 1B in the interfacial region between the two liquids and to the resulting high effective thermal conductivity due to mixing in that thin region at the molecular level, a profound rate of heat transfer between the two liquids would be attained; far above the rate established between the water and the cold wall in case 1A. The distribution of the temperature gradient in case 1B in both liquids and in the interface region between them is represented in Fig. 6. It is consistent and in accordance with the temperature

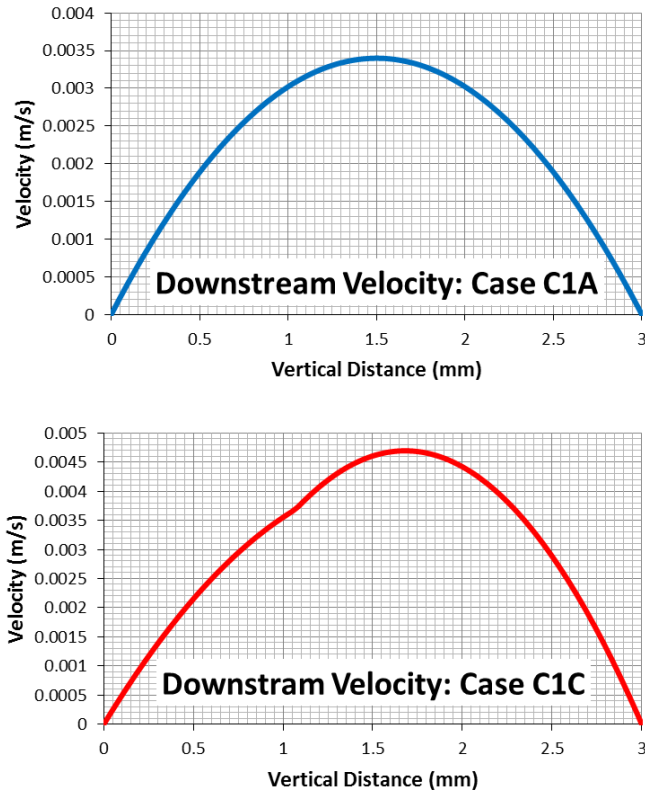


Figure 4: Downstream velocity profile at a downstream location 10 mm from the channel inlet

distribution presented in Fig. 5.

Table 6 gives a comparison between cases 1A and 1B in regards to the total amount of heat removed from the water, by the time the water passes through two downstream locations in the channel (10 mm away from the inlet and at the channel outlet). The percentages presented in table 6 are based on the initial total thermal energy input coming with the water at the channel inlet.

In Case 1B, the total heat removed from water in the first 10 mm of the channel length is about 100 percent more than that removed in case 1A. In case 1B the water loses about 61 percent of its initial total thermal energy by the time it leaves the channel. This percentage is about 34 percent in case 1A.

The above findings based on the numerical analysis, as will be elaborated on more in what follows, are consistent with the evidence extracted from the experimental

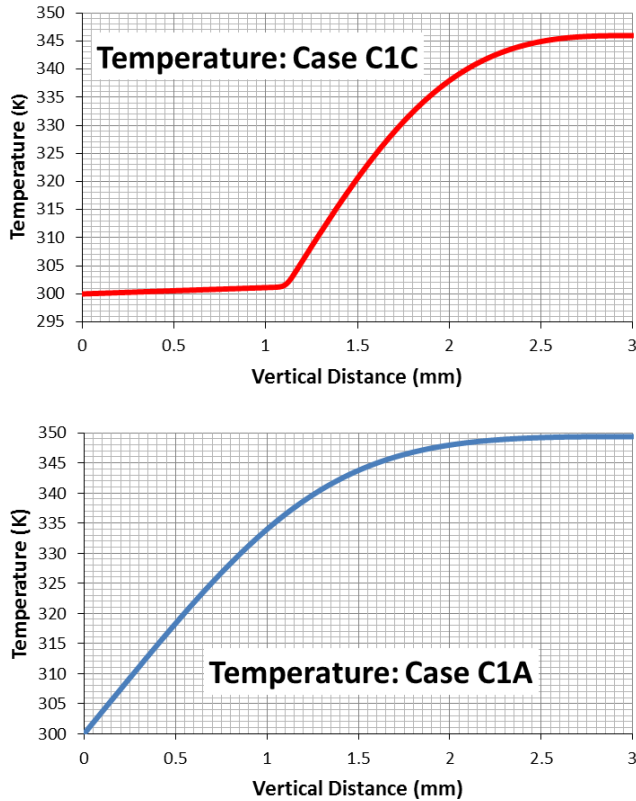


Figure 5: Temperature profile at a downstream location 10 mm from the inlet

results presented above. It should be recalled that the conditions and interface dynamics between the two liquids in the experiments conducted (gallium and water) might somewhat be different from their counterparts in the numerical runs. In the experiments the bulk of the two liquids was stationary (non-flowing), while in the numerical computations the two liquids were flowing in the channel, and in case 1B they flow with a relative speed between them. Nevertheless, it is still believed that in both cases the basic concepts underlying the overall phenomenon that is behind the enhancement in heat removal when the hot water is allowed to lose its heat while being in direct contact with the liquid gallium, are the same, especially after noting that in the channel flow considered in case 1B the two liquids flow at a relatively slow speed hence molecular dynamics is expected to be the dominant factor influencing the heat exchange between the two liquids; same as the expectation we would have based on the no-flow conditions prevailing in the experiments. Further

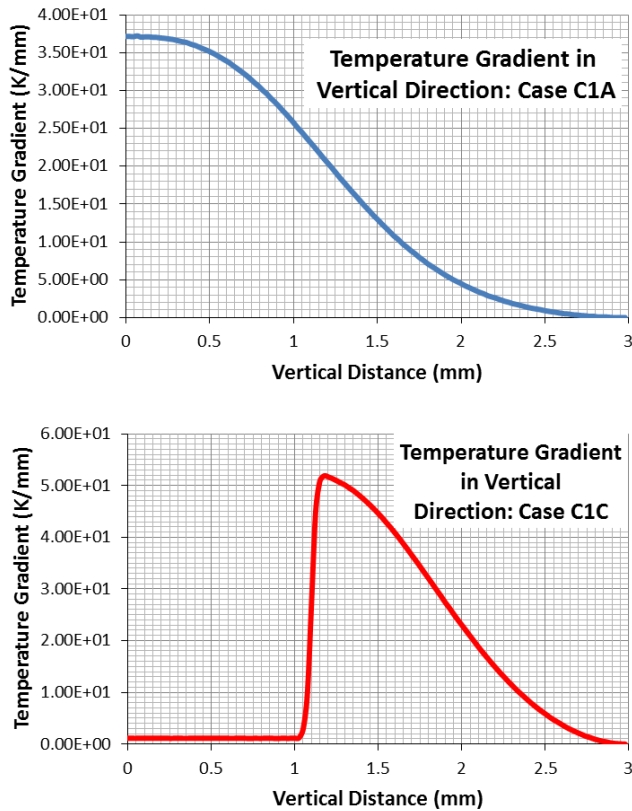


Figure 6: Temperature gradient in vertical direction at a downstream location 10 mm from the inlet

Table 6: Summary of the Heat Transfer Results for the Numerical Runs

Case No.	R=liquid metal/water ratio	Percent loss at x=10mm	Percent loss at channel outlet
1A	0	22%	34%
1B	2	45%	61%

elaboration on these aspects is under consideration by the present investigators in an on-going research project.

In what follows it is attempted to relate and discuss more the similarities and difference of the results of both the experimental part and the numerical simulations part. To have a unified basis for the analysis and discussions for both types of results, we consider the condition of heat loss from water at the channel outlet in the numerical simulations and the corresponding time in the experiments that was needed to achieve that same amount of heat loss in the experiments. Of interest here is the cooling time on unit mass basis (i.e. per unit mass of the cooled water) and the surface heat transfer area on unit mass basis, as well. These quantities would shed further light on the heat transfer process taking place both in the channel in the simulations and from the metallic cup in the experiments. Table 7 summarizes the above heat transfer-relevant quantities for all tests conducted, both experimentally and numerically, on unit cooled-water-mass basis.

Table 7a: Cooling times and heat transfer surface area characteristics in the numerical tests.

Test type and Number	Percent heat lost at channel outlet	Time needed to collect 40 grams of water at channel outlet (seconds)	Cooling surface area (m ² /gram of water)
1A: Numerical simulation	34%	5.88 ⇒ time per gram = 0.147 s	6.663 x 10 ⁻⁴
1C:Numerical Simulation	61%	5.88 ⇒ time per gram = 0.147 s	5 x 10 ⁻⁴

In Table 7 we consider two pairs of cases comparing and relating experiments to the numerical simulation results. In one hand, the first pair includes Test 1 and Test 1A, and on the other hand the second pair addresses Tests 3 and 1B. By considering the cooling time and cooling surface area ratios of the first pair (see Table 7), it may be concluded that the convective heat transfer coefficient in case 1A (although still comparable to that in case 1) is about 15 percent higher than that in case 1. Recall that dividing the time-ratio and the area-ratio for both pair of cases results in the ratio of the convective heat transfer coefficient of both cases in the pair, assuming the driving temperature difference for heat transfer between the two cases to be, more or less, comparable in the case of both pairs.

The slightly higher convection rate in case 1A (by about 20%) as compared with case 1, is attributed to the forced flow of the water in the channel in case 1A, as compared to the case of the still water in the cup in case 1 that just relies on

Table 7b: Cooling times and heat transfer surface area characteristics in the experimental tests.

Test Type and Number		Cooling surface area (m ² /gram of water)
#1: Exp. (no gallium)	Time for 40 grams to lose 34% of their initial heat = 92.5 s ⇒ time per gram = 2.31 s	4.91 x 10 ⁻⁵
#1 Exp. (no gallium)	Time for 40 grams to lose 61% of their initial heat = 208 s ⇒ time per gram = 5.2 s	4.91 x 10 ⁻⁵
#3: Exp. (with Gall.)	Time for 40 grams to lose 34% of their initial heat = 55.8 s ⇒ time per gram = 1.4 s	4.91 x 10 ⁻⁵
# 3: Exp. (with Gall.)	Time for 40 grams to lose = 61% of initial heat = 130.1 s ⇒ time per gram = 3.25 s	4.91 x 10 ⁻⁵

Table 7c: Comparisons of the cooling times, the heat transfer areas, and the convective heat transfer coefficients in the different tests.

Comparison of times in the different tests	Comparison of heat transfer surface area in the different tests	Comparison of convective heat transfer coefficients
Time to lose 34% heat in Test 1 / Time needed in Test 1A = 15.7 s	Area per gram in Test 1A / Area in Test 1 = 13.6	Conv. Heat Transfer Coefficient in Case 1A/ Conv. Heat Transf. Coefficient in case 1 = 1.2
Time to lose 61% heat in Test 3 / Time needed in Test 1C = 22.1	Area per gram in Test 1C / Area in Test 3 = 10.2	Conv. Heat Transfer Coefficient in Case 1C/ Conv. Heat Transf. Coefficient in case 3 = 2.16

conduction and some additional natural convection currents due to slight changes in density as a result of the variation in water temperature during the cooling process. Of course, under conditions of further increase in the speed of the water in the channel (i.e. under higher water flow rates) we still expect a larger difference in convective heat transfer coefficient between the cup case and the channel flows case, beyond the 20 percent given above.

On the other hand, in the second pair when gallium is used to aid heat transfer (in both cases 3 and 1B), the convective heat transfer coefficient in the case of the

channel flow (case 1B) is found (by dividing the time-ratio and the area-ratio in table 7) to be greater than the natural convective heat transfer effects in the cup case (case 3) by about 120 percent. This is a bigger difference between the channel and cup cases, than the 20 percent difference found above in the case of the first pair with no gallium.

To explain the above further, we may recall that in case 1B both water and gallium are flowing in the channel at the same time side by side, and hence part of the channel space (about one third) is occupied by gallium. This, for the same mass flow rate of water, would mean a faster flow of water in case 1B than in case 1A. This by itself may mean a higher convective heat transfer rate in case 1B than in case 1A, but still to have a higher convective heat transfer coefficient in case 1B that is by 120 percent more than in case 3, is more than to be explained by just the above effect. This leads us to conclude a bigger role of the presence of the interfacial dynamics due to the liquid metal (gallium) while flowing alongside with the hotter water and the resulting high temperature gradients on both sides of the interface between the two liquids. Further testing and experimentations of the above is still in progress and may be addressed in future publications by the authors. A final point in this regard to be mentioned is related to the results reported in a recent publication (Al Omari, S.-A. B., 2012), in which different numerical discretization schemes were used, which show that a second order discretization scheme for the numerical problem of the flow of the two liquids considered in this study would lead to a little bit less heat transfer rates than the ones reported above for the numerical computations, which in this study are based on a first order discretization scheme. For example, based on the second order discretization scheme the water in the above channel would lose about 55 percent of its initial heat by the time it exits the channel; compare this with the 61 percent loss considered above. Taking this into consideration, may give further insight into the better match between the heat transfer rates attained in the cup in the experiments and the channel flow in the computations. Also, this may lessen the role of the heat transfer by convection in the channel flow than what has been concluded above regarding the quantitative differences between the cup experiments and the numerical heat transfer results of the channel. Further consideration of the above findings will be tackled in an on-going project.

4.1 Mesh-Independence Testing of the Numerical Results

To show that the numerical results obtained are mesh independent, results using two different grids were obtained. In the first relatively rough Mesh (Grid 1), with a number of nodes of computational cells of 48,000 was utilized. A finer mesh (Grid 2) with double that number of cells (i.e. 192,000 cells) was used. Sample result that

shows the results obtained based on these two grids, are presented in Fig. 7 which shows the outlet temperature distribution based on both grids for two cases (case 1B and another case not presented in this paper for another liquid metal; mercury). As can be seen from both Figures, the results obtained are clearly mesh-independent. The results for the other computed quantities were also mesh-independent.

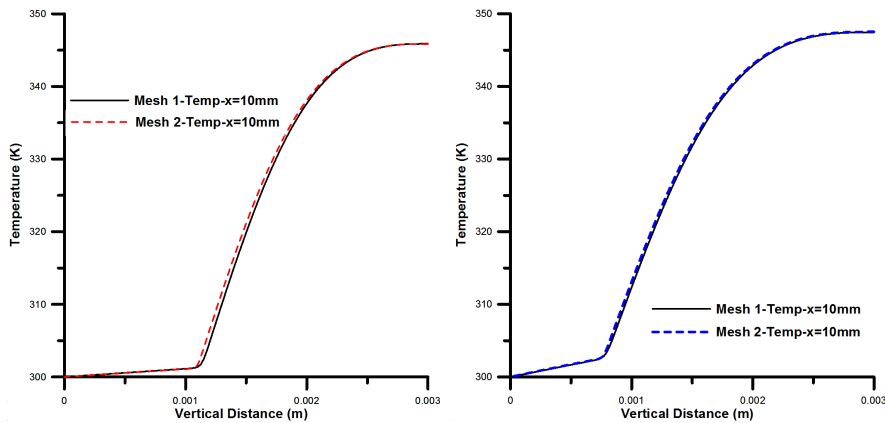


Figure 7: Mesh-sensitivity tests: Temperature distribution at channel outlet for two different cases

5 Conclusions

Heat transfer between hot water and colder liquid metal (liquid gallium) has been studied experimentally. In the conducted experiments the hot liquid water was allowed to exchange heat with liquid gallium in a metallic cup under solely natural convection heat transfer conditions. The bottom base of the cup was allowed to touch the surface of a cold water bath that is at approximately room temperature that acts as a thermal sink in which most of the heat transferred from the hot water through liquid gallium would eventually be dumped.

The experimental results were compared and related to some numerical simulations results for the heat transfer between hot water and gallium in a channel flow. Both the numerical simulations and the experiments verify clear enhancement in heat transfer rates when the high thermal conductivity liquid exists in direct contact with the hot water during the cooling process.

In the numerical simulation cases reported above, the superiority of the heat transfer rate aided by gallium over that without gallium is by about 80 percent (case 1B vs case 1A). The experiments above, on the other hand, show that gallium leads to

faster heat transfer from the hot water by about 60 percent, as compared with the case with no gallium (case 3 vs case 1). These two percentages are to some extent comparable, especially after considering the nature of the flow and heat transfer in both approaches, being purely natural convection in the cup experiments and one associated with some forced convection effects in the numerical simulations.

Compared to the heat exchange rates in the cup experiments, in the numerical simulations of the channel flow the heat transfer rate is faster by about 20 percent when water alone was present (case 1A vs case 1) and by about 120 percent when both water and gallium co-existed during the process (case 1B vs case 3).

The somewhat small difference (20 percent) in the case when only water is considered (Tests 1 and 1A), highlights that the heat exchange surface area was found to be the most dominant factor regarding heat transfer rates, and that forced convection effects are not that prominent in enhancing heat transfer significantly in the channel flows under the slow flow conditions studied. In the flow tests with both water and gallium were co-flowing in direct contact, the effect of natural convection was more clear due to the somewhat higher flow speeds of water were the space occupied by gallium in the channel would enforce the water (under the same steady flow conditions) to flow at faster speeds. However, after considering second order discretization schemes in the simulations (as suggested in Al Omari S.-A. B., 2012 the differences between the experimentally obtained results and the computed ones become less, hence undermining somewhat forced convection effects.

Acknowledgment

The financial support from the research affairs of UAE University to conduct the research reported is highly appreciated. The authors also acknowledge the help of the lab Engineer in the Department of Mechanical Engineering/UAEU Mr. Mohamed El-Said Hassanein and our research assistant student Mr. Maan N. Al Zareer for their help in the process of the preparation for the experiments.

References

- Al Omari S.-A. B.** (2011): Enhancement of heat transfer from hot water by co-flowing it with mercury in a mini-channel. *Int. Communication in Heat and Mass Transfer*, vol.38, no.8, pp.1073-1079.
- Al Omari S.-A. B.** (2011): Characteristics of the flow and heat transfer of gallium in a mini-scale channel under the effect of magnetic field. *J. Enhanced Heat Transfer*, vol.18, no.6, p.537-546.
- Al Omari, S.-A. B.** (2012): A numerical study on the use of liquid metals (gallium and mercury) as agents to enhance heat transfer from hot water in a co-flow

mini-channel system. *Heat Mass Transfer*, vol.48, no.10, pp.1735-1744. DOI 10.1007/s00231-012-1015-9,

Al Omari S.-A. B.; Haik, Y.; Abu Jdayil, B. (2010): Flow characteristics of gallium in a meso-scale channel under the influence of magnetic fields. *Int. Communication in Heat and Mass Transfer*, vol.37, no.8, pp.1127-1134.

Bar-Cohen, A.; Kraus, A. D.; Davidson, S. F. (1983): Thermal frontiers in the design and packaging of microelectronic equipment. *Mechanical Engineering*, vol.105, no.6, pp.53-59.

Chomdee, S.; Kiatsiriroat, T. (2006): Enhancement of air cooling in staggered array of electrical modules by integrating delta winglet vortex generators. *Int. Communications in Heat and Mass Transfer*, vol.33, pp.618-626.

Dumont, G.; Fontaine Vive Roux, Ph.; Righini, B. (2000): Water-cooled electronics. *Nuclear Instruments and Methods in Physics Research A*, vol.440, pp.213-223.

Incropera, F. P. (1988): Convection heat transfer in electronic equipment cooling. *ASME Journal of Heat Transfer*, vol.110, pp.1097-1111.

Incropera, F. P. (1999): Liquid Cooling of Electronic Devices by Single-Phase Convection. *John Wiley & Sons*.

Kwang-Soo, K.; Myong-Hee, W.; Jong-Wook, K.; Byung-Joon, B. (2003): Heat Pipe Cooling Technology for Desktop PC CPU. *Applied Thermal Engineering*, vol.23, pp.1137-1144.

Kothe, D. B.; Rider, W. J. (1995): Comments on Modeling Interfacial Flows with Volume-of-Fluid Methods. *Los Alamos National Laboratory Report*, 65P05, 76T05.

Ma K.-Q.; Liu, J. (2007): Nano Liquid-metal Fluid as Ultimate Coolant. *Physics Letters A*, vol.361, pp.252-256.

Naphon, P.; Wiriyaart, S. (2009): Liquid cooling in the mini-rectangular fin heat sink with and without thermoelectric for CPU. *International Communications in Heat and Mass Transfer*, vol.36, pp.166-171.

Nguyen, C. T.; Roy, G.; Gauthier, C.; Galanis, N. (2007): Heat transfer enhancement using Al₂O₃-water nanofluid for an electronic liquid cooling system. *Applied Thermal Engineering*, vol.27, pp.1501-1506.

Sathe, S.; Sammakia, B. (1998): A review of recent developments in some practical aspects of air-cooled electronic packages. *ASME Journal of Heat Transfer*, vol.120, pp.830-839.

The CFD package Fluent 6.3.

Tsai, T.-E.; Wu, H.-H.; Chang, C.-C.; Chen, S.-L. (2010): Two-phase closed thermosyphon vapor-chamber system for electronic cooling. *Int. Communications in Heat and Mass Transfer*, vol.37, pp.484-489.

Tang, H.; Wrobel, L. C.; Fan, Z. (2004): Tracking of immiscible interfaces in a multiple-material mixing process. *Computational Materials Science*, vol.29, pp.103-118.

Wits, W. W.; Vaneker, T.H.J.; Mannak, J.H.; Legtenberg, R. (2009): Novel cooling strategy for electronic packages: Directly injected cooling. *CIRP Journal of Manufacturing Science and Technology*, vol.1, pp.142-147.

Yeh, L. T. (1995): Review of heat transfer technologies in electronic equipment. *Journal of Electronic Packaging*, vol.117, pp.333-339.

