REPUBLIC OF TURKEY

YILDIZ TECHNICAL UNIVERSITY

GRADUATE SCHOOL OF SCIENCE AND ENGINEERING

INVESTIGATION ON SINGLE-PHASE HEAT TRANSFER ENHANCEMENT THROUGH MINIATURE PIN-FIN SURFACE EXTENSION FOR BATTERY THERMAL MANAGEMENT SYSTEMS

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MASTER OF SCIENCE THESIS

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September, 2023

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PİL TERMAL YÖNETİM SİSTEMLERİ İÇİN TEK FAZLI ISI TRANSFERİNİN MİNYATÜR PİN KANATLI YÜZEYLER İLE ARTTIRILMASINA YÖNELİK ARAŞTIRILMASI

A thesis submitted by Şafak ÜRKMEZ in partial fulfillment of the requirements for the degree of MASTER OF SCIENCE is approved by the committee on 19.09.2023 in Department of Mechanical Engineering, Mechanical Engineering Program.

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Signature

This study was supported by Yildiz Technical University Scientific Research Projects Coordination Department, Grant No: FYL-2022-4879

Dedicated to my family and

my advisor

ACKNOWLEDGEMENTS

First of all, I am very grateful to our YTU BAP office for their contribution to make this project a reality. Special thanks to my advisor Assoc. Prof. Dr. Zafer GEMİCİ, Assoc. Prof. Dr. Mete BUDAKLI, Mehmed Sinan KÖMEK, Sami MALKI, Ahmet GÜNEY and my family for their support.

Şafak ÜRKMEZ

TABLE OF CONTENTS

LIST OF SYMBOLS

A	Area
°C	Degree Celsius
cm	Centimeter
c_p	The Specific Heat Capacity
d	The Density of Water
D_h	The Characteristic Length
F	Friction Coefficient (Darcy)
h	The Height of the Channel
h	The Convection Heat Transfer Coefficient
$\bar{h}$	The Average Convection Heat Transfer Coefficient
H	The Height of the Fin
I	Current
k_c	The Thermal Conductivity Coefficient of Copper
k_w	The Thermal Conductivity Coefficient of Water
K	Kelvin
l	The Length of the Fin
L	The Length of the Heat Sink
m	Meter
mm	Millimeter
$\dot{m}$	The Mass Flow Rate
N_f	The Number of Fins in the Finned Structure
Nu	Nusselt Number
P	Power
Pa	Pascal
Pr	Prandtl
Q	Heat
q_{cond}	The Heat Conduction per Unit Area
R	The Total Thermal Resistance
R	Function
Re	Reynolds Number
t	The Thickness of the Fin
T	Temperature

T_C	The Corrected Thermocouple Value
T_T	The Thermocouple Value
u	The Uncertainty
u_{x_e}	The Uncertainty of an Equipment
u_{x_r}	The Uncertainty of Random Component
V	Volt
v	Water Flow Rate
w	Water
w	The Width of the Heat Sink (Copper Plates)
W	Watt
x	Distance
Δ	Delta
ΔP	The Pressure Difference of the Heat Sink
ΔT_{ln}	The Logarithmic Mean Temperature Difference
η	The Efficiency of the Fins
η^*	The Total Efficiency of the Fins

LIST OF ABBREVIATIONS

bot	Bottom
AC	Alternating Current
DC	Direct Current
et al.	Et Alia
etc.	Et Cetera
EV	Electric Vehicle
EVs	Electric Vehicles
FOM	Figure of Merit
HVAC	Heating, Ventilating and Air Conditioning
in	Inlet
Li-ion	Lithium-ion
PEC	Performance Evaluation Criteria
PCM	Phase Change Material
PPRC	Polypropylene Random Copolymer
RTD	Resistance Temperature Detector
R&D	Research and Development
TMS	Thermal Management System
tot	Total

LIST OF FIGURES

LIST OF TABLES

ABSTRACT

Investigation on Single-phase Heat Transfer Enhancement Through Miniature Pin-fin Surface Extension for Battery Thermal Management Systems

Şafak ÜRKMEZ

Department of Mechanical Engineering

Master of Science Thesis

Supervisor: Assoc. Prof. Dr. Zafer GEMİCİ

Today, heat exchangers are used in every field, from the white goods industry to the automotive industry, and they are very important. Nowadays, when global warming comes to the fore, the importance of energy efficiency is increasing. In this thesis, miniature heat exchangers are studied and in order to develop them, experiments have been made and the results have been given. This thesis is mainly defined for a heat exchanger that can be used to cool batteries in electric cars, but the results of this thesis can also be used in other miniature heat exchangers. Fully developed flow is provided at the inlet of the heat sink, a small gap is left between the fin tips and the top surface of the channel, experiments are performed with water at high Reynolds numbers and their effect on heat transfer and pressure drop is studied. To the best of the author's knowledge, there is no such study in the literature.

The thesis consists of 6 main chapters. It starts with the introduction where similar studies in the literature are reviewed and the parts of the thesis that differ from the literature are explained and the aims of the thesis are given. In the second part, the experimental setup was explained and all the equipment was presented in detail. The third part explains how the experiments were carried out and the formulas used

to calculate the test results. In the fourth part, the uncertainty analysis, which is very important for experimental studies, is included. In the fifth part, the results obtained from these experiments and calculations are given and examined. In the sixth part, the results obtained from the experiment are briefly summarized and what can be done next is mentioned.

Keywords: battery cooling, rectangular pin fin, tip clearance, miniature heat exchanger.

ÖZET

Pil Termal Yönetim Sistemleri İçin Tek Fazlı Isı Transferinin Minyatür Pin Kanatlı Yüzeyler ile Arttırılmasına Yönelik Araştırılması

Şafak ÜRKMEZ

Makine Mühendisliği Anabilim Dalı

Yüksek Lisans Tezi

Danışman: Doç. Dr. Zafer GEMİCİ

Isı değiştiriciler günümüzde beyaz eşya sanayisinden otomobil sanayisine her alanda kullanılmaktadır ve önem arz etmektedir. Küresel ısınmanın ön plana çıktığı şu günlerde enerji verimliliğinin de önemi gittikçe artmaktadır. Bu tezde minyatür ısı değiştiriciler incelenecek olup, deneyler yapılmış ve sonuçlara yer verilmiştir. Bu tez daha çok elektrikli arabalarda bataryaların soğutulmasında kullanılabilecek bir ısı değiştiricisi için tanımlanmıştır ama bu tezden elde edilecek çıktılar başka minyatür ısı değiştiricilerinde de kullanılabilir. Buradaki ısı alıcısının girişinde tam gelişmiş akış sağlanmıştır, ısı değiştiricisinin içindeki kanatlarla üst yüzey arasında ufak bir açıklık bırakılmıştır, yüksek Reynolds sayılarında suyla deneyler yapılmıştır ve bunların ısı transferine ve basınç düşümüne etkisi incelenmiştir. Böyle bir çalışma literatürde neredeyse hiç yapılmamıştır.

Tez çalışması 6 ana bölüm içermektedir. İlk başta giriş kısmıyla başlamaktadır, burada literatürdeki benzer makaleler incelenmiştir ve tezin literatürden farklı olduğu kısımlar ve amaçları verilmiştir. İkinci kısımdaysa deney düzeneği anlatılıp tüm ekipmanlar detaylı olarak tanıtılmıştır. Üçüncü kısımdaysa, deneylerin nasıl yapıldığı ilk başta anlatılmış ve sonrasında deney sonuçlarını hesaplamak için

kullanılan formüller bahsedilmiştir. Dördüncü kısımda, deneysel çalışmalar için çok önemli olan belirsizlik analizine yer verilmiştir. Beşinci kısımda bu deneylerden ve hesaplamalardan elde edilen sonuçlara yer verilip bu sonuçlar irdelenmiştir. Altıncı kısımdaysa deneyden ulaşılan çıktılar kısaca özetlenip bundan sonra yapılabileceklerden bahsedilmiştir.

Anahtar Kelimeler: bataryaların soğutulması, dikdörtgen fin kanat, uç açıklığı, minyatür ısı değiştiricisi

1
INTRODUCTION

Nowadays, energy consumption is increasing drastically, and the amount of energy consumption is one of the signs of civilization development according to Michio Kaku [1]. As our human civilization develops, our increasing energy demand and consumption is a normal fact, by the way, more developed countries tend to consume more energy as well. It is common knowledge that energy is crucial and indispensable for humanity. In the increasing trend of energy consumption, energy efficiency becomes more and more important, because every energy consumption has some effect on our environment, and it mostly be a harmful way. Air-sea-river-soil pollution and global warming are examples of these harmful effects. Therefore, most of the countries try to improve energy efficiency at the expense of cost increase or R&D investment (setting regulations on household appliances is an example of this). As the importance of energy efficiency is high for our world, this master thesis topic was selected for increasing energy efficiency or making a way for increasing energy efficiency by developing heat exchangers.

Heat exchangers are used in so many places, such as in the refrigerator, dryer, air conditioner, some of the power generation equipment, cars, and in HVAC systems. Therefore, any improvement in heat exchangers will affect so many places that its impact will be high. Although heat exchangers are widely used, there are still gaps in the literature and this thesis aims to contribute to fill the gap. This heat sink is dimensionally suitable for the cooling of batteries, and the use of batteries has started to be widespread, especially in the automotive industry. For all the above reasons, this study was conducted.

1.1 Literature Review

Sen Hu and Keith E. Herold [2] stated that models derived from air studies (Colburn factor) overestimate the performance of liquids. For this reason, experimental studies with air are not discussed extensively in this literature review. The literature review begins with the most similar articles, which are rectangular miniature channels using pin fins and water as the coolant. After that, battery cooling methods and articles are explained in the second part of the literature review because this miniature heat exchanger can be applied to batteries. Finally, some of the studies using air as a coolant are told.

1.1.1 Most Similar Studies

Thermal management of the electro-mechanical devices and the batteries are very important topics and using pin-fin in heat sinks is one of the prominent methods for setting the temperature of these devices to the required temperatures. Duangthongsuk et al. [3]states that using the pin fin heat exchanger is one of the best thermal management technics. So, in this title, first, studies with pin-fin geometries will be given and then, a few studies related to plate-pin-fins and plate-fins will be given as another usage methods of providing thermal management.

Liang et al. [4] conducted experiments on a rectangular narrow channel with no fins which has a 1.8 mm height and find that friction coefficient is getting smaller with increasing Reynolds Number. Furthermore, there is no significant difference in terms of heat transfer whether the experimental setup is placed horizontally or vertically and the results are nearly same when experiments conducted in steady state and in transient. Also in the rectangular duct, the Nusselt number is larger in laminar flow but smaller in turbulent flow than in the circular duct. In another study performed by Liang et al. [5], the height of the rectangular channel was changed to 1 mm and 2.5 mm and it is stated that Nusselt number is greater in both laminar and turbulent flow in rectangular channels than in circular channels, on the other hand friction coefficients are higher than in circular channels. Also, correlations of friction factor and Nusselt number are given in both of the studies performed by Liang et al. [4], [5]. Wang et al. [6] worked on transition region in rectangular channel to obtain critical Reynolds number with changing Prandtl numbers and found that with decreasing Prandtl number, lower and upper critical Reynolds numbers increase. The other output of their experiments is friction factor increases with increasing Prandtl numbers for a steady Reynolds number and for transition region. Wang et al. [7] conducted tests in narrow channel for isothermal and non-isothermal conditions and found relations between the friction factor-Reynolds numbers and Nusselt-Reynolds numbers, also correlations were given.

Aliabadi et al. [8] carried out experiments with 8 different pin fins, water as a working fluid and between 100-900 Reynolds, the results showed that half-circular miniature pin fin heat sink has the biggest heat transfer coefficient, on the other hand it has the most pressure drop, too. To find optimum results between these terms, the performance evaluation criterion was utilized, and the circular pin fin was found the best. Thermal resistance was found as another performance criterion for electronic devices [9] and half-circular showed the best result [8], also convective resistance has the largest proportion between capacitive and conductive resistance. In addition to all these, 2 correlations were derived which give Nusselt number and friction factor of different shapes of pin fins with a deviation less than ±5%. Duangthongsuk and Wongwises [10] used a miniature circular pin-fin inline arrangement and a zigzag flow channel with a single cross-cut structure in their experiments and utilized thermal resistance for comparing their hydrothermal performances and found that the zigzag flow channel with a single cross-cut structure has better performance than the miniature circular pin-fin inline arrangement. Nevertheless, an inline arrangement is known to have worse performance than a staggered arrangement, so it should be done that try to compare these with using a staggered arrangement in miniature fins and use other performance factors besides the thermal resistance. In another study of Duangthongsuk and Wongwises [11], they examined only heat transfer coefficient between 500 and 4000 Reynolds numbers, Nusselt number or friction factor or hydrothermal performance were not given and water was used as a coolant. Duangthongsuk and Wongwises have another study [12] in which they compared miniature circular pin fin heat sink and miniature square pin fin heat sink. The results show that circular pin fins have 6-9% better performance than square pin fin. Bahiraei et al. [13] conducted tests in miniature pin fin heat sinks using three different pin fins which are circular, triangular and drop-shaped pin fins and compared them using thermal resistance. Results show that the best of them (having the least thermal resistance) was circular, then drop-shaped pin fin. The results also show that with increasing velocity, thermal resistance also increases. Daeseong et al. [14] generated correlations for the narrow rectangular channel with and without the entrance region and compared them with other generated correlations. These correlations can be used in calculating the hydrothermal performance of fins because some of the given formulas don't include any fin structure. Moore [15] studied the tip clearance of fins and found that as the tip clearance of fins increases, the pressure drop decreases as expected. Also, the friction factor decreases with increasing Reynolds Numbers.

There is another type of heat transfer enhancement technique which is called plate-fin heat sinks and can be used together with pin fins. However, although there are studies in the literature, it's not known which is better for hydrothermal performance [16]. Yu et al. [17] studied on plate-pin-fin heat sinks and found that it is also a good method to increase heat transfer, but states that the pressure drop is a little high. To eliminate this disadvantage, thin straight splitter rear pin-fins were used by some researchers [16], [18], [19]. Hosseinirad et al. [16] conducted an experimental study on plate-fin heat sinks with a Reynolds number range of 50 to 250 and water as the working fluid. They tried different types of splitters adjacent behind and in front of the pin fins and found that one of the performance criteria decreases with Reynolds number, while another performance criterion remains almost the same. It could be said that among the different types, the arched splitter with forward arrangement showed the best performance in this range of Reynolds numbers.

Khoshvaght-Aliabadi et al. [20] conducted experiments to obtain the thermal-hydraulic performance factors of different plate fin channels using water as the working fluid and the results show that the pin fin channel has the lowest thermal-hydraulic performance due to its highest pressure drop. It was also determined another performance criterion which is called the possible reduction of surface area. The results are nearly the same as with the thermal-hydraulic performance factors. This article shows that different types of fins have superior properties compared to pin fins. Similar results were reported in another article by Khoshvaght-Aliabadi et al. [21]. The tested plate-fins involve plain, perforated, offset strip, louvered, wavy, vortex-generator, and pin channels. According to three performance criteria, the vortex generator, wavy and offset strip have better performance than the pin channel, so this article also indicates that there may be other types that are better than the pin channel. In the study [22], Khoshvaght-Aliabadi took a channel type which is vortex generator (it was the one that has the best thermal-hydraulic performance) and tried to find the optimum shape, dimension and angle of the wing and dimension of the channel for water, oil and ethylene glycol at laminar flow. Chamanroy et al. [23] investigated the effects of straight and wavy miniature heat sinks with straight and wavy pin fins on heat transfer and pressure drop performance. In this study, 'thermal resistance' and 'heat transfer rate/pumping power' were used as hydrothermal performance factors, which differed from the above studies. The results show that as the Reynolds number increases, both performance factors decrease (Reynolds numbers up to nearly 600). According to these factors, straight miniature heat sinks with straight pin fin and wavy miniature heat sink with wavy pin fin had the best performances. The results also show that the conductive resistance remains almost the same at different Reynolds numbers; convective and capacitive resistances decrease as expected.

Khoshvaght-Aliabadi et al. [24] also made another experiment with water in miniature heat sinks. They used 6 different straight and wavy miniature heat sinks and the best hydrothermal performance factors were the wavy interrupted-staggered miniature heat sink and the straight interrupted-staggered miniature heat sink. The straight interrupted-staggered miniature heat sink is almost the same as the one studied in this thesis and it shows that choosing this type of heat sink for use is logical.

1.1.2 Studies Related with Battery Thermal Management

Batteries have been increasingly popular with the growing trend of electric vehicles, and battery thermal management is a very important task for the reasons of affecting its performance and safety. The proposed heat-exchanger in this study can be used for cooling batteries, therefore this part of the literature review should be done. Li-ion batteries are mostly investigated in these studies because now it has the best properties for EVs. It will be better starting with a review paper:

Arora [25] discussed existing and emerging technologies for battery cooling but started with the effect of temperature on the Li-ion battery and the heating of the battery. At high temperatures, the battery ages faster and, more importantly, its safety is compromised. At about 85 °C, the cell begins to melt and the chemical reactions become self-sustaining, which is called thermal runaway. Therefore, the fire of the Li-ion battery is difficult to extinguish. Cold is also detrimental to batteries. It decreases the performance of the batteries a lot. The range of the 2012 Nissan Leaf decreases from 138 miles to 63 miles at -10 °C. Although the best temperature range for the Li-ion battery varies in the literature, most of the range is similar. In this study, the range of this temperature is taken between 15-40 °C as can be seen in detail in Figure 1.1.

Battery cell temperature	Cause	Leads to	Effect
High	Electrolyte decomposition	Irreversible lithium loss	Capacity fade
	Continuous side reactions at low rate	Impedance Rise	Power fade
	Decrease of accessible anode surface for Li-ion intercalation		
	Decomposition of binder	Loss of mechanical stability	Capacity fade
25 °C – 40 °C	Maximum cycle life		
15 °C – 24 °C	Superior energy Storage capacity		
Low	Lithium plating	Irreversible loss of lithium	Capacity/ power fade
	Electrolyte decomposition	Electrolyte loss	

Figure 1.1 Temperature ranges and how the Li-ion battery is affected by temperature changes [25]

Preheating is important in cold temperatures, and there are several methods in the literature. One of them is to use the impedance which is relevant for ohmic heat generation and the other way is to use the internal resistance for temperature rise. Placing heaters on both sides of the battery and a fluid heating strategy are other types of battery preheating techniques, but Pesaran et al. [26], [27] say that the most efficient way emerges as a core heating of the battery in their study. Conventional battery cooling techniques are classified in three ways, as shown in Figure 1.2.

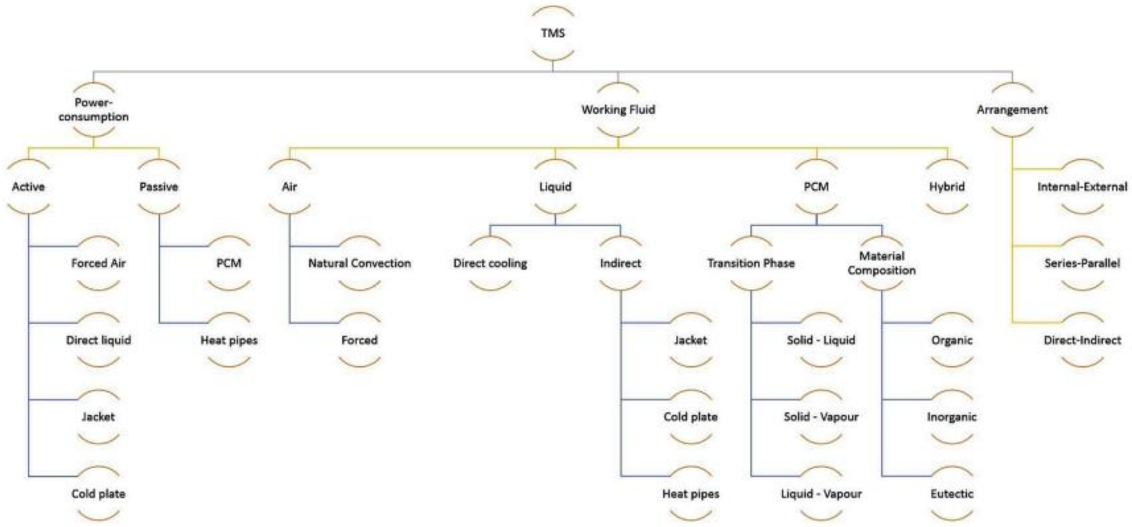


Figure 1.2 Temperature classification of conventional battery cooling in three ways [25]

The most reasonable classification of TMS is the working fluid, and these are air-cooled, liquid-cooled, phase change materials (PCMs) and any combination of these. Air is the simplest way to cool the battery, it's also cheap and requires little maintenance. The other attractive features of air cooling are its low cost, low manufacturing cost, no liquid leakage, and no additional weight [28], [29]. On the other hand, some studies show that under harsh conditions, the air is insufficient to cool the batteries in EVs [30]–[36]. Therefore, air cooling systems can be coupled with other cooling technologies such as PCM, thermoelectric coolers, etc., and can overcome harsh conditions. For these reasons, some of the EV cars use air cooling systems and some don't [25]. A detailed summary of air cooling of batteries in EVs and hybrid EVs is recommended in the study [31]. With the use of liquid, battery cooling is easy to handle and the parasitic load (the energy used to keep the battery temperature in the required range) is very low compared to air cooling, but leakage is a serious problem. It also requires a lot of maintenance, it is complex, and can be subject to fouling and scaling which can occur in tubes or channels that need to be cleaned for good heat transfer. Direct liquid cooling uses a dielectric liquid (such as deionized water and silicon-based oil) and liquids in direct contact with the batteries. In indirect liquid cooling, the liquid flows through the jacket or tubes around the batteries. A good comparison of the cooling techniques mentioned so far (air cooling, indirect and direct liquid cooling) is reported in the study by Chen et al.

[37]. A CFD study was performed for a battery cell, taking into account the technics, the cooling fluid and its flow characteristics for EVs'. Conclusion: air cooling can't provide sufficient heat dissipation with respect to the other cooling techniques. Direct cooling and indirect cooling are the best techniques to reduce the temperature of the battery out of liquid, fin and air cooling. Fin cooling was mentioned which adds too much extra weight, but the more efficient type of fin cooling (like pin fin) could be used. Heat pipes are also another technique for cooling the batteries [25]. Heat pipes use latent heat for heat exchange by changing the state of the fluid (liquid to gas and gas to liquid) and this way is more effective than conduction heat transfer. Nevertheless, heat pipes aren't used much in EVs because of high capital cost, but new improvements in manufacturing technologies in aluminum can replace copper heat pipes, so it can reduce the capital cost and weight of the heat pipes, and the use of heat pipes in EVs can be increased.

One of the emerging cooling technologies is thermoelectric cooler and it has many advantages: light, no noise and moving parts, low price. Besides these advantages, its cooling is inefficient [38] compared with other battery cooling techniques, if the performance of the thermoelectric cooler is improved, it can be the best cooling technique [25]. Thermoelectric acoustic cooling, magnetic cooling and internal cooling are also promising ways to cool the battery. The best cooling system will be a combination of these systems, because all of them have different disadvantages and advantages, a combination of them can overcome these disadvantages. Shi et al. [39] state that the energy demand of air cooling and liquid cooling can reach 40% of the battery energy of the battery pack. Therefore, the importance of the passive thermal management system is high, or more durable batteries or batteries that produce less heat are needed.

Arora et al. [40] presented a promising battery thermal management system. In this study, they placed PCMs between the batteries, but the batteries are placed upside down, and this way was shown to increase the temperature uniformity of the pack, which is very important for battery packs. For normal battery cell has terminals at the top of it and first melt the top of the PCM, so the top of the battery is much hotter than the bottom. On the other hand, in the inverted position of the batteries, the PCM melts mostly on the bottom side, but with density difference, solid and liquid parts of the PCM mix and uniform temperature occurs. Moreover, on the top of the PCM, there is a thermoelectric generator to take heat and both cool the PCM and generate energy from the heat, so the parasitic power is reduced, which is used for the battery thermal management system.

1.1.3 Studies Related with Heat Sinks Using Air as a Coolant

Şara et al. (2003) used staggered pin fins with a clearance at the top of the fins in a rectangular duct. In this study, air was used as the working fluid and the fins were square in shape. They showed that staggered arrangement is better than inline arrangement in heat transfer, but the inline arrangement is better in pressure drop or friction factor. They also concluded that the average Nusselt number and friction factor increased with decreasing clearance ratio [41].

Peles et al. (2005) showed that thermal resistance increases with increasing Reynolds number. Thermal resistance is one of the criterions for measuring the efficiency of a heat exchanger [42].

Vatanparast et al. (2020) used numerical techniques for rectangular staggered pin fins to compare entropy generation according to Reynolds number (50-75-100). They found that entropy generation increases with increasing Reynolds number. In addition, the total entropy generation increases with increasing fin width [43].

Tanda et al. (2001) studied the heat transfer and pressure drop in a rectangular channel using air as the working fluid and diamond-shaped fins as the extended surface. Under the same conditions, the Nusselt number of the staggered arrangement is always higher than the in-line arrangement of the fins, and the pressure drop is higher in the staggered arrangement. The best thermal performance was obtained with low intermediate fin densities. Also, the thermal performance of diamond-shaped elements showed that the heat transfer rate was increased by a factor of up to 4.4 for the same mass flow rate [44].

Dhumne and Farkade et al. (2013) studied pressure drop and heat transfer in perforated fins in a staggered arrangement. They found that perforated fins have a higher Nusselt number than solid fins, and as the number of fins increases, the heat transfer increases. Even at high Reynolds numbers, the friction factor of perforated fins is slightly better than that of solid fins. Above 10000 Reynolds number, the low

Reynolds number flow shows better performance than the higher Reynolds number in pin fins [45].

One of the other studies related to these topics is also mentioned in the study [46]. It is similar to the previous studies, but it concludes differently that the efficiency of the in-line fins is slightly higher than that of the staggered fins. Similar studies have been done for the fluid material which is air [47], [48].

1.2 Objective of the Thesis

This work has been done to contribute to science and to improve the heat exchangers/heat sinks by increasing the heat transfer and decreasing the pressure drop. There are too many parameters that affect the performance of a heat exchanger and there have been too many studies on this subject, but there is still room to improve its performance. Nowadays, the importance of energy efficiency is increasing because climate change is one of the biggest challenges in the world and so reducing the energy consumption will directly reduce the carbon footprint since the world's energy is mostly produced from fossil fuels. So, this thesis was conducted to improve a battery heat exchanger and thus contribute to the reduction of the carbon footprint of the world.

1.3 Hypothesis

There are many studies in the literature using water with nanoparticles, and air, but only a few studies using only water as the working fluid. Also, most of the studies have been done at low Reynolds numbers, but here high Reynolds numbers are used. Furthermore, this study investigates the heat transfer and pressure drop in a heat sink with a clearance between the fin tip and the top surface of the rectangular channel. There is a study in the literature that is partially similar, but it does not include performance factors that show the change in performance with each change in working fluid temperature and flow rate. This heat exchanger is expected to provide higher heat transfer and a performance factor greater than one. In this study, an experimental system was set up and experimental studies were performed for rectangular fins at different temperatures, heat loads, and flow rates. The results are compared with the literature.

2
EXPERIMENTAL SYSTEM

The system in which the heat exchanger is tested consists of a rectangular aluminum channel, PPRC pipes, 4 pumps, a datalogger, a power supply, a computer, a U-type manometer, a flow meter, a chiller and a water tank. An avometer was also used to measure the resistance of the electric cartridge heaters. The test apparatus is shown in Figure 2.1.

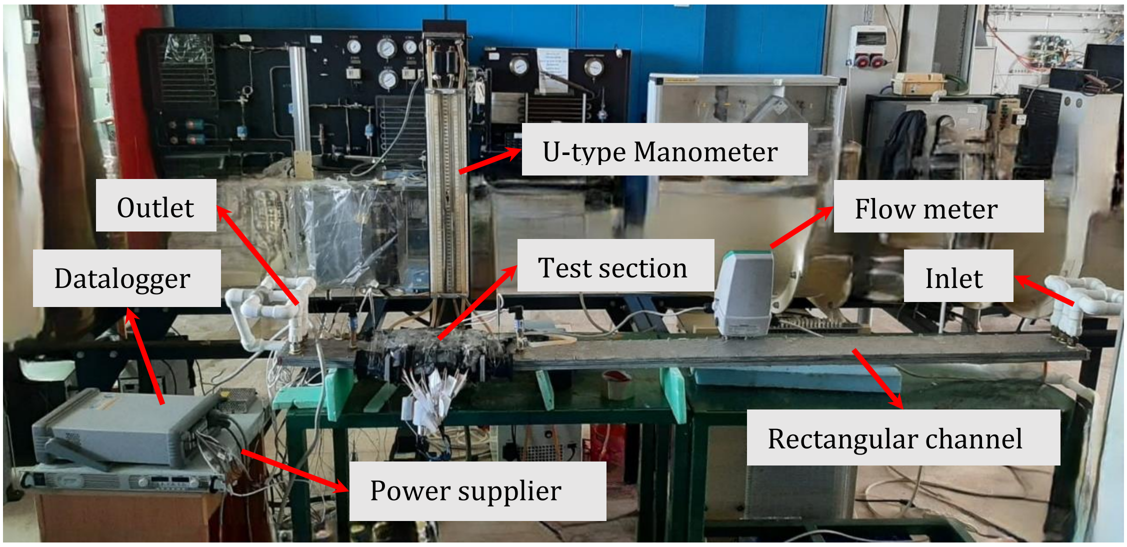


Figure 2.1 The test apparatus

Water is used as the working fluid that flows through the system. The chiller sets the temperature of the water, and it keeps the temperature of the water at 10-20-30 °C. 2 pumps take the water from the chiller and deliver it to the water tank where the circulating water at the heat exchanger system mixes with the water coming from the chiller. Another 2 pumps are located under the water tank and flood the water into the system, providing the necessary power and pressure. Then the water reaches the rectangular channel through the PPRC pipes. On the PPRC pipes before the entrance of the channel, there is a slide valve that adjusts the desired flow rate and Reynolds number in the heat exchanger. The right part of the rectangular channel (the part before the test zone) is intentionally long enough (1.5 m) to achieve a fully developed flow. This long rectangular narrow channel is connected to the heat sink (test zone). The heat sink has 3 parts: finned structure (made of copper, can be seen in Figure 2.2, water flows through this structure), 14 cartridge electric heating resistors (one of them is shown in Figure 2.3) located inside the aluminum rectangular wall that wraps all around the finned structure.

Figure 2.2 Finned structure

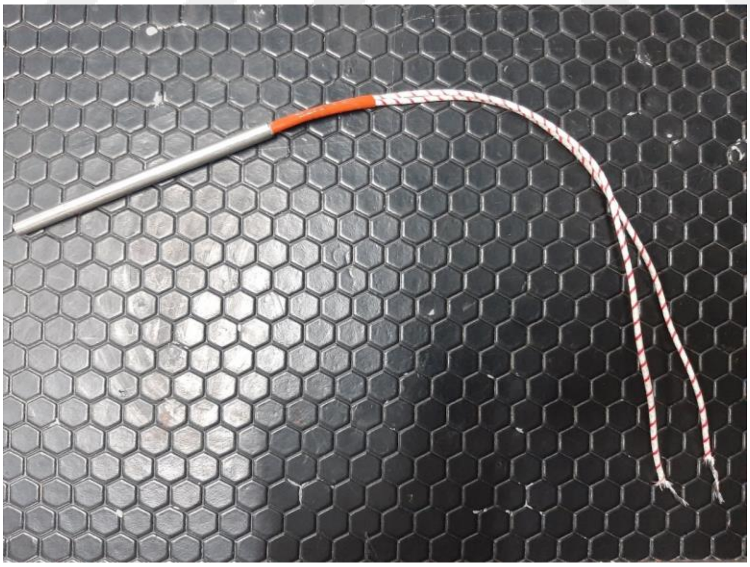


Figure 2.3 One of the cartridge electric heating resistors

The system can be described in 2 circuits (in each circuit there are 2 different pumps): the first one is from the chiller to the water tank and the second one is from the water tank to the heat exchanger system. The pumps of the first circuit are used only for the transfer of water from the chiller to the water tank, the transfer of cooled or heated water in the chiller to the water tank. The second circuit begins at the water tank. Water flows from the water tank to the system inlet by means of 2 pumps, just before the water reaches the system inlet there is the slide valve. The water inlet to the system is where the long rectangular channel begins. After the water reaches the entrance, it goes through the long rectangular channel, then the heat sink and then the short rectangular channel. After that it comes back to the water tank with the help of PPRC pipes on which there is also a flow meter.

In this system, the electric heaters add heat to the circulating water in two ways: from both the top and the bottom surfaces of the heat sink, and from the bottom surface only. By adjusting the Reynolds numbers between 4000-16000 with a valve, it was studied how these changes affect the heat transfer and pressure drop. In the finned structure, the finned copper surface was placed on the bottom of the heat sink and the plain copper without fins was placed on the top of the heat sink.

The 3-dimensional model of the system (the small rectangular channel, the heat sink and the long rectangular channel, respectively) is shown in Figure 2.4.

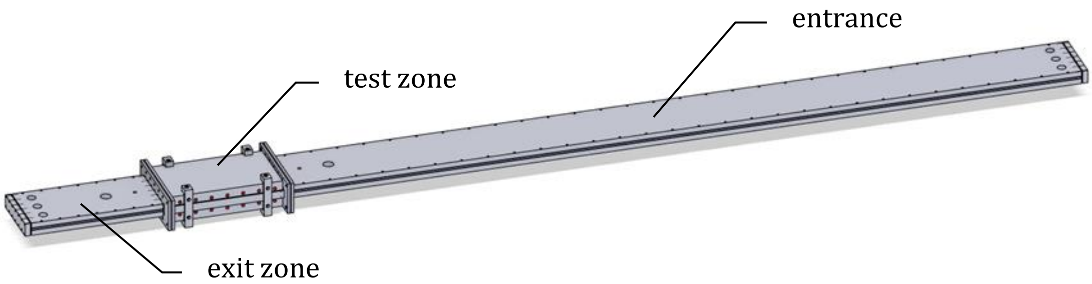


Figure 2.4 3-dimensional model of the system

The heat sink and channels contain 6 thermocouples and 2 resistance temperature detectors (RTDs). RTDs measure temperature more accurately than thermocouples. 2 RTDs are used in the inlet and outlet of the heat sink to measure the inlet and outlet temperatures of the water. There is one thermocouple at the top and bottom of the heat sink and the heat exchanger contains 4 thermocouples placed on the left and right sides of the heat exchanger. The diagram for the thermocouple locations is shown in Figure 2.5. In Figure 2.5, the right part is the long channel where the water enters and the left part is the short channel where the water exits. Numbers 15 and 14 indicate RTDs, while 10, 11, 12, 16, 17, 18 indicate thermocouples. 12 and 16 numbered thermocouples measure the temperature outside the rectangular aluminum wall surfaces (these two aren't used in the calculations or anywhere, just out of control). 10, 11, 17 and 18 numbered thermocouples are located inside the finned structure (copper parts) but not in contact with water.

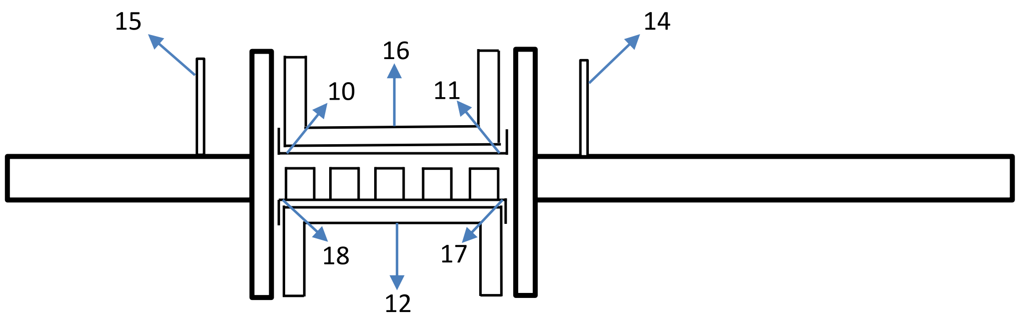


Figure 2.5 The thermocouple and RTD locations

There are two ports before and after the heat sink that are connected to the U-manometer with plastic tubes to measure the pressure drop across the heat sink. In a U-manometer, a high liquid level difference is needed to reduce the error in the visual reading. For this reason, at low Reynolds numbers, meaning low pressure drops, water was used instead of mercury because at low Reynolds numbers, the pressure difference is small, so the difference between the mercury levels becomes small and the uncertainty increases. Therefore, for low Reynolds numbers water was used and for high Reynolds numbers mercury was used to measure the pressure difference between the inlet and outlet of the heat sink by using U-manometer.

In this system, the whole channel is made of aluminum, except for the finned structure (it is made of copper). The inner diameter of the plastic tubes is 7 mm, PPRC pipes is 17 mm and the cross section of the rectangular aluminum channel is 5 x 100 mm x mm. All the thermocouples and RTDs are connected to the data logger, which collects data and transfers it to the computer.

The copper plates have an area of 214 x 100 mm x mm (w x L) (Figure 2.6). The height of the channel (h) is 5 mm, and the dimensions of the fins are 1 x 10 x 4.9 mm (t x l x H) (Figure 2.7). The clearance between the fin tip and the upper surface of the finned structure is 0.1 mm. Therefore, the fin height is 4.9 mm. The other geometric dimensions are given in the cross section of the heat sink and rectangular fin configuration shown in Figure 2.8.

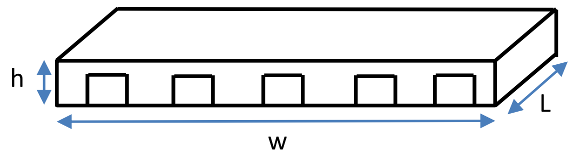


Figure 2.6 The dimensions of the structure with fin

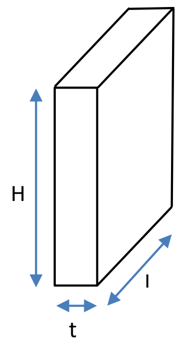


Figure 2.7 The dimensions of the one fin

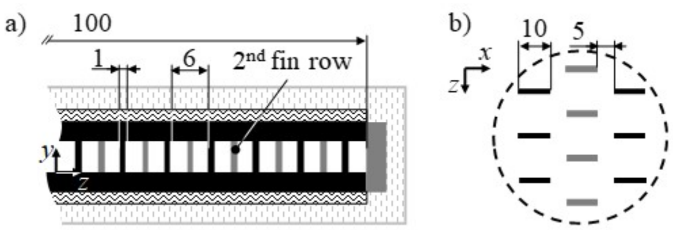
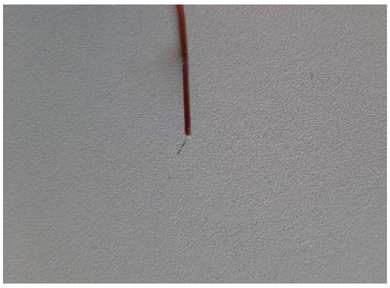


Figure 2.8 a) Cross section of the heat sink, b) Configuration of rectangular fins [49]

Details, photos, and accuracies of each equipment are given in Table 2.1. In order to perform the uncertainty analysis, the accuracies were given and the uncertainty analysis is performed in the later sections. The thermocouples and RTDs were connected to a data logger (Keysight 34970A - 34901A multiplexer) and calibrated by a reference instrument (Fluke) with an accuracy of ±0.2 K.

Parameters	Equipment	Accuracies	Photos
Volume Flow Rate	Volume Flow Meter (SITRANS F M MAGFLO MAG5000 - SIEMENS)	± 0.4%	
Pressure	U-Type Manometer	± 0.5% FS	
Data Collection Ports	Datalogger (Keysight 34970A – 34901A multiplexer)	-	

Pressurizer of Water	2 Pumps	-	
Pressurizer of Water	2 Pumps	-	
Volt-Resistance-Ampere meter	Avometer (M3004)	For Voltage: ± 0.5% (DC) ± 1% (AC) For Current: ± 1.5% For Resistance: ± 0.7%	
Temperature of Fins	T - type thermocouple	± 0.2 K	

Inlet and Outlet Water Temperature	RTD	± 0.1 K	
Current	Keysight N5767A Power Supply, 1500 W	± 0.1%	
Voltage	Keysight N5767A Power Supply, 1500 W	± 0.1%	
Length	Millimeter scale	± 0.5 mm	

Height	Vernier caliper	± 0.001 mm	
Width	Vernier caliper	± 0.001 mm	

Table 2.1 Each equipment details and photos, calibration ranges and errors

The experimental system includes a filter between the water tank and the slide valve, just before the water enters the long aluminum rectangular channel, to keep unwanted materials from the system. Nevertheless, the used chiller is a little bit old and the filter can't keep the rust from the water, hence rusted water keeps going into the heat exchanger system and unfortunately the results can be affected a bit by this rust.

Meanwhile, the heat sink section was completely insulated. Thus, the calculations assume that all the given heat from the cartridge heaters is transmitted to the water and not to the environment. The RTD used at the exit of the heat sink is not included in the assessment of the performance of the heat sink because it increases the uncertainty too much, that's why the outside of the heat sink is fully insulated and all of the given heat is assumed to be transferred to the water.

Cartridge heaters have 60 Volt, 145 Watt characteristics and their resistances were evaluated by avometer (each has almost the same ohmic value) and the upper part of the heat exchanger is a total of 4.3 ohms and the lower part is 4.2 ohms. Thus, it is assumed that the lower and upper parts of the heat sink give the same heat to the water.

To better understand the experimental system, the diagram of the system is shown in Figure 2.9.

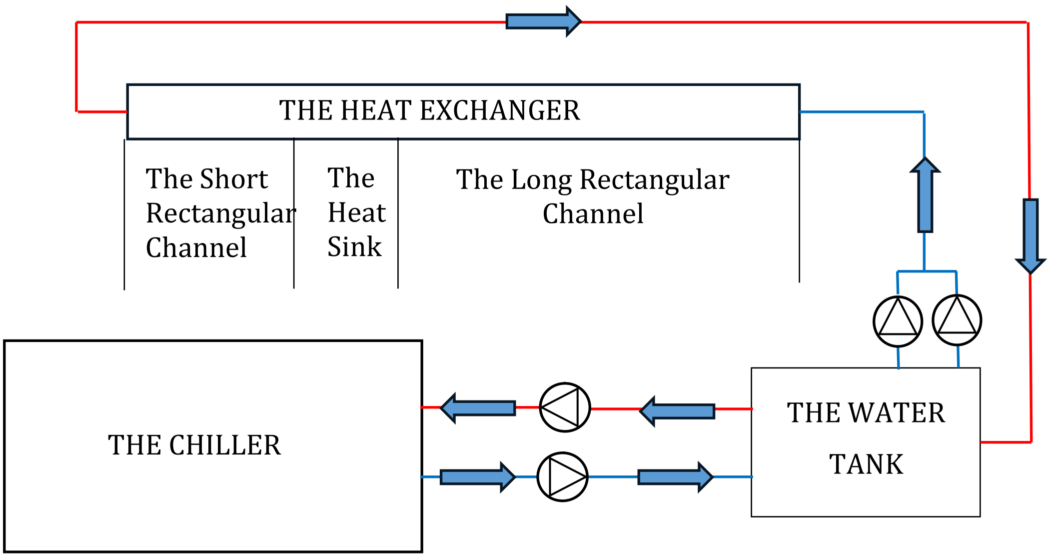


Figure 2.9 The diagram of the experimental system

3
EVALUATION OF THE EXPERIMENTS AND EXPERIMENTAL METHODOLOGY

3.1 Evaluation of the Experiments

Experiments were conducted in different types of flow rates, temperatures, heat given parts, and heat powers. While the Reynolds numbers are set to 4000, 6000, 8000, 10000 and 12000 for 10 °C, 4000, 6000, 8000, 10000, 12000, 14000 and 16000 for both 20 °C and 30 °C. Higher Reynolds numbers can be achieved in the same system when the temperature is increased because Reynolds number includes density and viscosity and these are affected by temperature. Above, all the experiments were done in the same way as for the state where heat is given at both top and bottom and for the state where heat is given only at the bottom part of the heat sink.

The duration of each experiment is at least 15 minutes and each experiment has been started when the system has reached nearly steady state conditions (this can be observed by looking at temperatures). Each water inlet temperature is in the interval of ±1 °C, for example, if an experiment is said to be conducted at 10 °C, the water inlet temperature is between 9 and 11 °C at any time of any experiment. After the experiments are performed, the data are exported to Excel and temperature-time graphs are drawn, and the most stable temperature interval is selected and data are recorded there. Flow rates are read every 2.5 minutes and averaged the whole data in each experiment. Now the flow rates are almost constant in this time interval. To get more accurate results, this method was chosen. The pressure difference was measured at the beginning and at the end and averaged. These pressure differences also almost never change during this period.

3.2 Experimental Methodology

Nusselt number and friction factor are the dimensionless numbers used as appropriate means to compare heat transfer rate and pressure drop. Because they are unitless, they can be used to compare different heat exchangers and heat sinks which have different characteristics. In this thesis, the Nusselt number will be found by finding the convection heat transfer coefficient (3.1) and the Nusselt-Reynolds graph will be found to compare heat transfer. Friction factor-Reynolds graph will also be found to compare pressure drop.

$$Nu = \frac{h.D_h}{k_w} \tag{3.1}$$

In this equation: h is the convection heat transfer coefficient, D_h is the characteristic length, k_w is the thermal conductivity coefficient of water. To find the Nusselt number, the convection heat transfer coefficient should be evaluated first. Since there are two different heating modes of the system, namely heating from both top and bottom and heating from bottom only, the convection heat transfer coefficient should be determined separately for these two different conditions.

First, the method of calculating the heat transfer coefficient for bottom-only heating is explained. The given thermal power is equal to the given electrical energy per second from 'Keysight N5767A Power Supply', which is found by multiplying volt and ampere (3.2) displayed on the instrument screen. T_{out} is then calculated from the equation (3.3). In this equation, inlet water temperature given in the equation (3.4) is read from the RTD and $\dot{m}$ and c_p are the water properties. After that, the heat transfer coefficient can be deduced from the equation (3.5). In this formula, the logarithmic mean temperature difference (3.6) is taken to be more consistent and $T_{bot,in}$ is the corrected value of thermocouple 17 and $T_{bot,out}$ is the corrected value of thermocouple 18 (thermocouples are shown in Figure 2.5). The bottom area consists of fin area and base area, can be calculated by the following equations (3.7), (3.8) and (3.9), and includes the geometric parameters such as H, t, l, w and L. N_f is the number of fins in the finned structure. Also η^* (3.10) is the total efficiency of the fins and includes the efficiency of the fins (η) (3.11). m changes with $\bar{h}_{bot}$ in the equation (3.12), so this makes Q_b equation implicit and $\bar{h}_{bot}$ can be evaluated iteratively. k_c is the thermal conductivity coefficient of copper.

$$P = V.I \tag{3.2}$$

$$Q = Q_{bot} = \dot{m}.c_p.\Delta\mathrm{T} \tag{3.3}$$

$$\Delta T = T_{in} - T_{out} \tag{3.4}$$

$$Q_b = A_b . \bar{h}_{bot} . \eta^* . \Delta T_{lnbot} \tag{3.5}$$

$$\Delta T_{lnbot} = \frac{\left(T_{bot,in} - T_{in}\right) - \left(T_{bot,out} - T_{out}\right)}{\ln\left(\frac{T_{bot,in} - T_{in}}{T_{bot,out} - T_{out}}\right)} \tag{3.6}$$

$$A_{bot} = A_{fin} + A_{base} \tag{3.7}$$

$$A_{fin} = N_f . (2\left(H + {}^{t}/_{2}\right). l + 2\left(H + {}^{t}/_{2}\right). t) \tag{3.8}$$

$$A_{base} = w . L - N_f . t . l \tag{3.9}$$

$$\eta^* = 1 - \frac{A_{fin} . (1 - \eta)}{A_{bot}} \tag{3.10}$$

$$\eta = \frac{\tanh\left(m\left(H + \frac{t}{2}\right)\right)}{m\left(H + \frac{t}{2}\right)} \tag{3.11}$$

$$m = \sqrt{\frac{\bar{h}_{bot} . P}{k_c . A}} = \sqrt{\frac{\bar{h}_{bot} . 2(l + t)}{k_c . l . t}} \tag{3.12}$$

Second, the method of calculating the heat transfer coefficient for both heating from the top and bottom is explained. The given heat is equal to the sum of these heatings, the given heat power is equal to the multiplication of volt and ampere (3.13) and T_{out} can be found by the formula (3.14). In this equation, inlet water temperature given in the equation (3.15) is read from the RTD and $\dot{m}$ and c_p are the water properties. The calculation of the $\bar{h}_{bot}$ is mentioned above and $\bar{h}_{top}$ is calculated from the equation (3.16). The logarithmic temperature difference (3.17) is taken to be more consistent and $T_{top,in}$ is the corrected value of thermocouple 11 and $T_{top,out}$ is the corrected value of thermocouple 10. The area is calculated from the equation (3.18).

$$P = V . I \tag{3.13}$$

$$Q = Q_{top} + Q_{bot} = \dot{m}.c_p.\Delta T \tag{3.14}$$

$$\Delta T = T_{in} - T_{out} \tag{3.15}$$

$$Q_t = A_t.\bar{h}_t.\Delta T_{lntop} \tag{3.16}$$

$$\Delta T_{lntop} = \frac{(T_{top,in} - T_{in}) - (T_{top,out} - T_{out})}{\ln\left(\frac{T_{top,in} - T_{in}}{T_{top,out} - T_{out}}\right)} \tag{3.17}$$

$$A_{top} = w.L \tag{3.18}$$

In these equations, the only unknown parameter is the convection heat transfer coefficient, so the average heat transfer coefficient is found from these equations and the average heat transfer coefficient is the same for the Q_{top} and Q_{bot} in heating from the top and bottom conditions.

Thermocouples 17, 18, 11 and 10 are located inside of the copper plates and there is 2 mm of copper between the thermocouples and the water. Thus, corrected thermocouple values ($T_{bot,in}$, $T_{bot,out}$, $T_{top,in}$, $T_{top,out}$) are obtained from Fourier's law (3.19).

$$T_C = T_T - \frac{x.q_{cond}}{k} \tag{3.19}$$

T_C is the corrected thermocouple value, T_T is the thermocouple value, x is the distance (2 mm) between the thermocouple and the water, q_{cond} is the heat conduction per unit area.

The friction factor (3.20) is calculated from the following equation. In this thesis, all of the used friction factors are Darcy friction factor, not Fanning friction factor.

$$f = \frac{2.\Delta P.D_h}{d.v^2.L} \tag{3.20}$$

ΔP is the pressure difference of the heat sink, d is the density of water, v is the flow rate of water and L is the length of the heat sink.

The total thermal resistance is a factor for finned surfaces to signify the heat transfer performance for electronic devices [9]. It consists of conductive, convective and capacitive resistances (3.21) [8].

$$R_{total} = R_{conductive} + R_{convective} + R_{capacitive} = \frac{T_{base} - T_{w,in}}{Q} \tag{3.21}$$

In this equation, T_{base} is the average of inlet and outlet temperatures of the heat given surface in the heat sink. $T_{w,in}$ is the water inlet temperature to the heat sink.

There are many performance evaluation criteria (PEC) for heat sinks in the literature [8], [16], [20], [23], [49], [50], but in this thesis, one of the most widely used is selected (3.22) [16], [49]. In some papers, it is also called Figure of Merit (FOM).

$$FOM = \frac{{Nu_{enhanced}}/{Nu_{smooth}}}{({f_{enhanced}}/{f_{smooth}})^{1/3}} \tag{3.22}$$

For Nu_{smooth} and f_{smooth}, there are too many correlations in the literature, some of them can be found in these studies [7],[51]. In this thesis, correlations from Cengel, Y.A. and Ghajar, A. [9] are used, which are Gnielinski (1976) equation for Nu_{smooth} (3.23) and for fully developed turbulent flow f_{smooth} (3.24).

$$Nu_{smooth} = \frac{\left(\frac{f_{smooth}}{8}\right)(Re - 1000)Pr}{1 + 12.7\left(\frac{f_{smooth}}{8}\right)^{0.5}\left(Pr^{\frac{2}{3}} - 1\right)} \tag{3.23}$$

$$f_{smooth} = 0.184Re^{-0.2} \tag{3.24}$$

4
UNCERTAINTY ANALYSIS

Uncertainty analysis is a very important and essential analysis for experimental systems [52]–[56]. Without this analysis, the test results cannot be trusted, and according to this analysis what accuracy of the equipment is needed can be defined, or if the experimental system has a high uncertainty, the decision of the more sensitive equipment should be selected, and the experiments can be repeated based on this decision. For example, if it is claimed that new fins in a heat sink improve heat transfer by 5% compared to the previous fins, but there is an uncertainty of ±6%, it is not possible to say that this test is valid. This means that the experiments did not show an improvement in heat transfer and should be repeated with more accurate and sensitive equipment. Another different example for uncertainty analysis to understand why this calculation is important is: which element this material made of is needed to be given. To do that, the density of the material can be measured as Archimedes suggested. If 2 researchers measure the density of the crown and decide whether it is 18-karat gold or alloy, which have 15.5 gram/cm^3 and 13.8 gram/cm^3 densities, respectively. One can find it is made of alloy with respect to measurements and one can find 18-karat gold due to the uncertainties of the devices. If uncertainty analyses aren't performed in experiments, it can lead to incorrect decisions or assumptions. Uncertainty analysis can be used to decide whether the claim is true or not, or whether the sensitivity and accuracy of the equipment meets the need to make that judgement.

To define the uncertainty in the calculations, the following equation for uncertainty by Kline and McClintock (1953) is used [57].

$$u_R = \sqrt{\left(\frac{\partial R}{\partial x_1}\right)^2 u_{x_1}^2 + \left(\frac{\partial R}{\partial x_2}\right)^2 u_{x_2}^2 + \cdots + \left(\frac{\partial R}{\partial x_n}\right)^2 u_{x_n}^2} \tag{4.1}$$

In this formula, u_R is the uncertainty of R, R is a function depending on the variables x_1, x_2,..., x_n and u_{x_1}, u_{x_2}, ..., u_{x_n} are the uncertainties of these variables. If a variable is measured in a certain amount in time arrival and the mean of it is taken, then to find the uncertainty of it (4.2): the standard deviation of these measured values is taken and the square of it is summed with the square of the equipment uncertainty. The square root of the final value gives us the uncertainty of the variable. If the calculations are estimated with 95% confidence, the instrument error should be divided by 2 at the beginning and the final result should be multiplied by 2 (4.3), because the uncertainty of random component (error from standard deviation) was estimated with 68% confidence.

$$u_{x_1} = \sqrt{u_{x_{1e}}^2 + u_{x_{1r}}^2} \tag{4.2}$$

In this equation, $u_{x_{1e}}$ is the uncertainty of the equipment and $u_{x_{1r}}$ is the uncertainty of random component.

$$u_{x_1} = 2\sqrt{\left(\frac{u_{x_{1e}}}{2}\right)^2 + u_{x_{1r}}^2} \tag{4.3}$$

Thus, (4.2) is estimated with 68% confidence and (4.3) is estimated with 95% confidence (the equipment uncertainty estimates in the equations should also be with 68% confidence in (4.2) and 95% confidence in (4.3)).

4.1 Uncertainty Analysis Results

In the uncertainty analysis, convective heat transfer coefficient (*h*), Nusselt number (*Nu*), Reynolds number (*Re*), pressure difference (ΔP), and friction factor (*f*) were calculated for the experiments. In each experiment, both the percentage of the uncertainties of *h* and *Nu* were the same, because in the Nusselt number formula, the most influential part for the uncertainty is *h*. In addition, the uncertainty of *h* and *Nu* increases with increasing *Re*, as expected, because the temperature difference between the thermocouples (inlet and outlet temperatures) decreases with increasing *Re*, and any decrease in the temperature difference increases the uncertainty. Temperature changes have almost no effect on the uncertainties of the *h* and *Nu*. The uncertainty percentages of the Reynolds numbers are the same for each experiment (0.4%). For sufficiently accurate readings at low pressure drops, water is used instead of mercury in the manometer to increase liquid level and reduce uncertainty. So, at the low Reynolds numbers the uncertainty is low because the water has a low density and this results in a high water level difference in the manometer, just as at high Reynolds numbers the mercury level difference in the manometer is high because of the high pressure drop. Consequently, the uncertainty is the highest at the mid Reynolds numbers, such as 6000 and 8000, where the pressure difference is measured with mercury and the level difference is relatively small. The highest uncertainties calculated from the experiments are shown in Table 4.1.

	Uncertainty Rate
h	6.9%
Nu	6.9%
Re	0.4%
P	2.1%
f	2.3%

Table 4.1 The maximum uncertainty rates of the experiment

5
RESULTS AND DISCUSSION

To understand the effects of Reynolds numbers, the way of heat supply, and the water inlet temperature on the heat transfer and the pressure drop: ΔT, Nu, ΔP, f, R_{tot} and the performance evaluation criterion are studied in relation to Reynolds numbers. Graphs showing the variation of these parameters with Reynolds number are plotted. In this way the optimum operating conditions of the heat sink can be determined.

5.1 Surface Temperature Differences

Only the temperature differences of the heat sink surfaces for the 'both from bottom and top' case are compared here. This is because during the 'from the bottom only' tests, the exit surface thermocouple had a reading error and it was not possible to calculate temperature differences. This evaluation is done to see how the temperature differences are realized under different conditions, because the temperature difference along the battery cell is required to be as small as possible for the sake of the battery's performance. For this reason, the difference between the outlet and inlet temperature of each top and bottom surface is taken. These are the differences of corrected values of the thermocouples 10-11 and 18-17. These temperature differences are shown in Figure 5.1 for 10-20-30 °C ($Pr_{average}$ = 9.44-6.94-5.4 respectively).

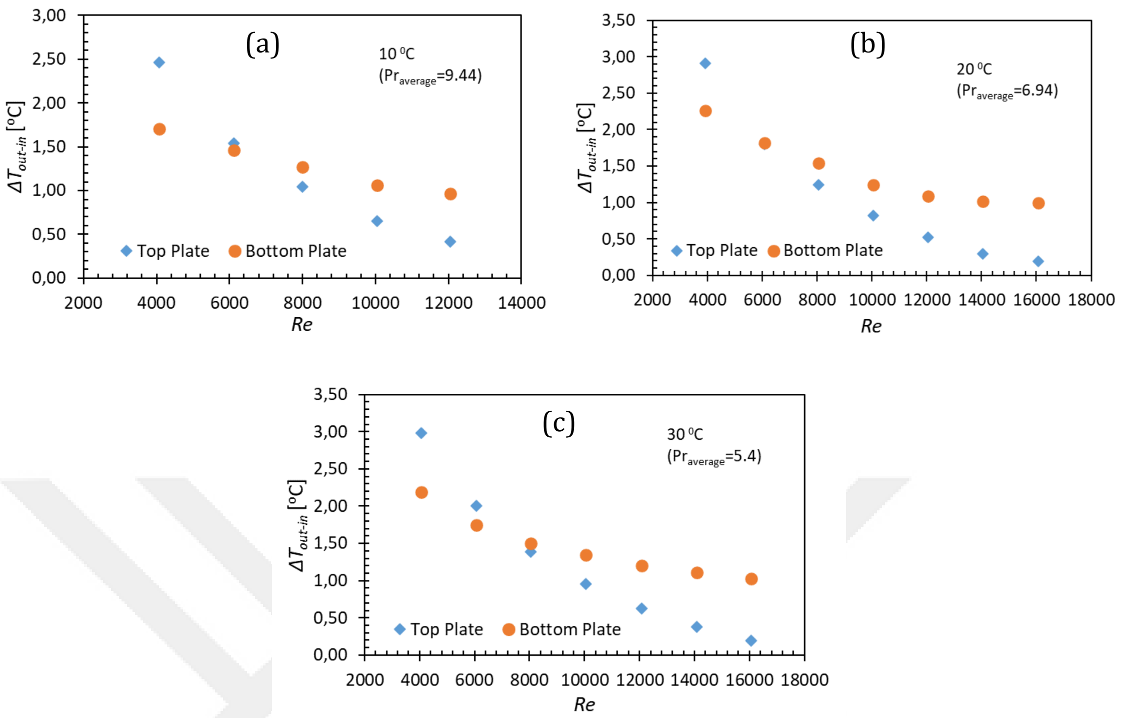


Figure 5.1 Temperature differences between inlet and outlet of the bottom and top plates for 'both from bottom and top' case at different water inlet temperatures: a) 10 °C, b) 20 °C, c) 30 °C

As can be seen in Figure 5.1, the temperature difference of the top plate (finless plate) is high at low *Re,* but low at high *Re*. This means that at low *Re*, the heat transfer enhancement with fins on the bottom plate works (because of the high heat transfer surface), but at high *Re*, the additional turbulence caused by the fin tip clearance becomes more important and the temperature difference becomes smaller, so that a more uniform temperature distribution can be obtained. These results show that at high *Re*, fin tip clearance can be used to obtain more uniform temperature distribution, which is very important for battery cooling.

5.2 Heat Transfer Coefficient

The convection heat transfer coefficient (*h*) is an important parameter for understanding the heat transfer rate in the heat sink. It is preferable to use the dimensionless form of the heat transfer coefficient, the Nusselt number. The results of *h* for different temperatures are given in Figure 5.2. The heat transfer coefficient increases as the temperature of the water decreases. Therefore, to increase the heat transfer, it was good to reduce the temperature of the liquid. It is also seen that the heat transfer coefficient increases as expected as the Reynolds number increases.

To understand how the way of heat supply affects the heat transfer coefficient, a comparison was made and the results are shown in Figure 5.3. According to this figure, the way of heat supply has almost no effect on *h* in the Reynolds range studied.

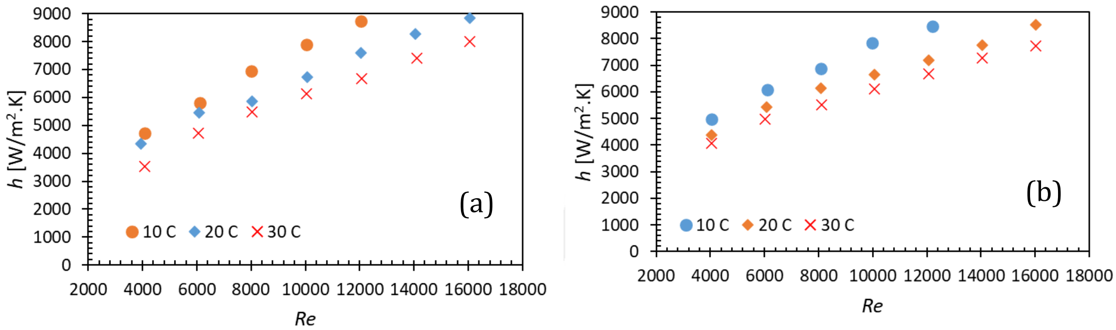


Figure 5.2 Heat transfer coefficient with respect to Reynolds number for different water temperatures: a) heat is given both from top and bottom b) heat is given only from bottom

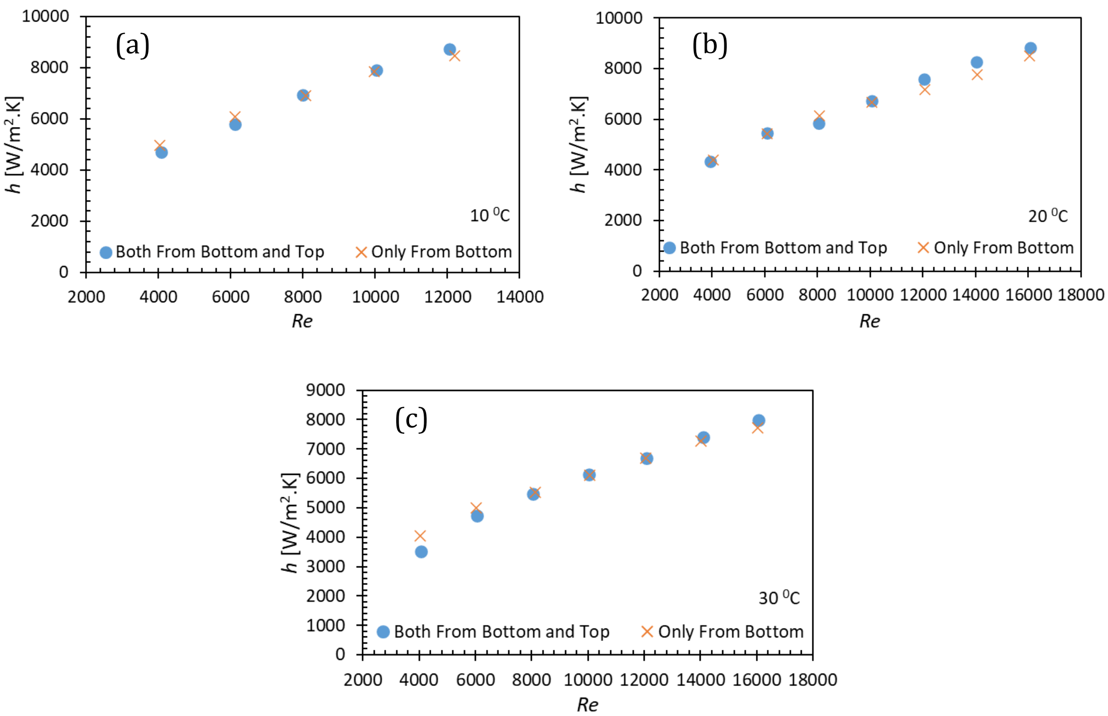


Figure 5.3 The changes of *h* with respect to *Re*, the way of heat supply and water temperature: a) 10 °C, b) 20 °C, c) 30 °C

5.3 Nusselt Number

The Nusselt number changes directly proportional to the convection coefficient and inversely proportional to the conduction coefficient. Since the conduction coefficient does not change much with temperature, the Nusselt number graphs are similar to the convection coefficient graphs, as expected. The *Nu* variations can be seen in Figure 5.4, Figure 5.5, and Figure 5.6. In the graphs, besides the variation of the Nusselt numbers of the finned geometry with Reynolds number at different inlet temperatures, the Nu values of the rectangular duct without fins, which we used for comparison in the calculation of Figure of Merit (FOM), are also given.

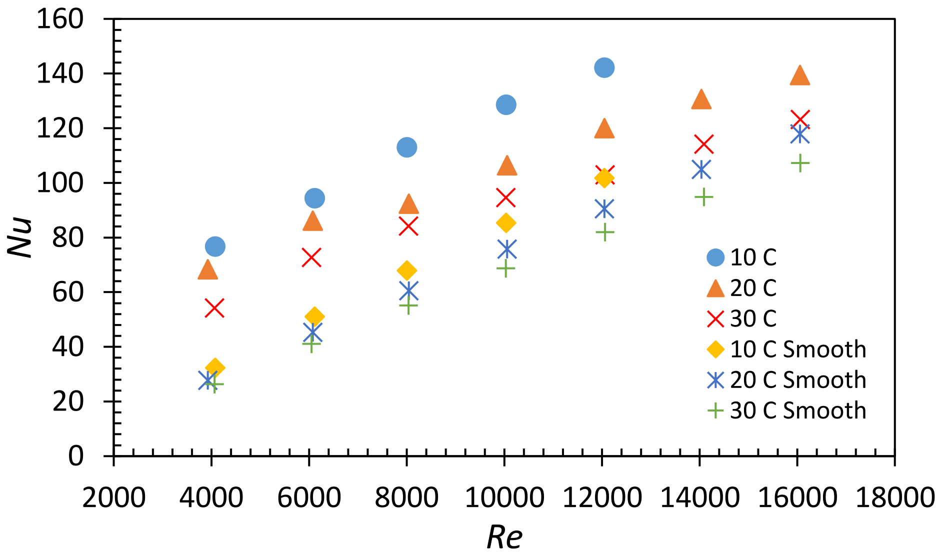


Figure 5.4 The Nusselt numbers in relation to Reynolds numbers and temperatures for finned and smooth rectangular channels at the case of 'both from top and bottom'

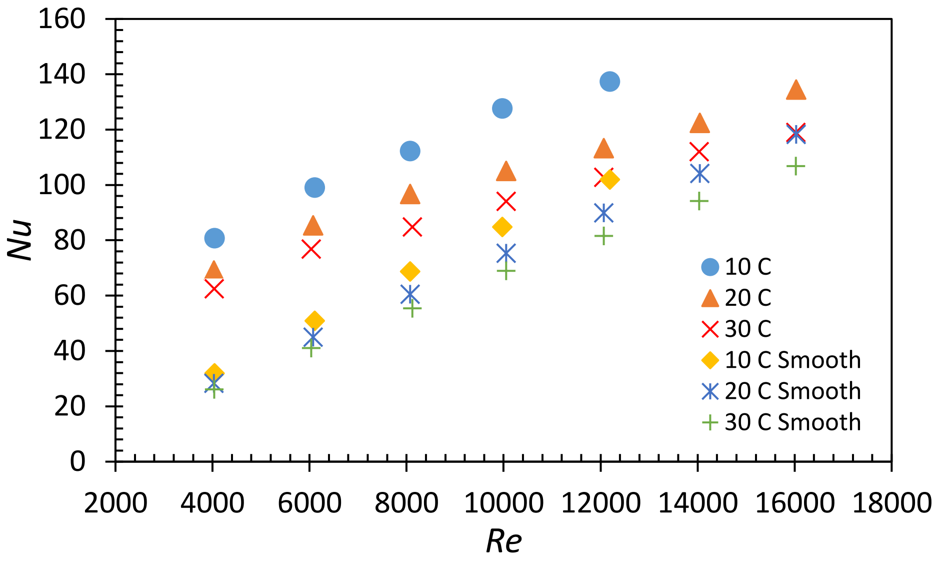


Figure 5.5 The Nusselt numbers in relation to Reynolds numbers and temperatures for finned (for the case of 'only from bottom') and smooth rectangular channels

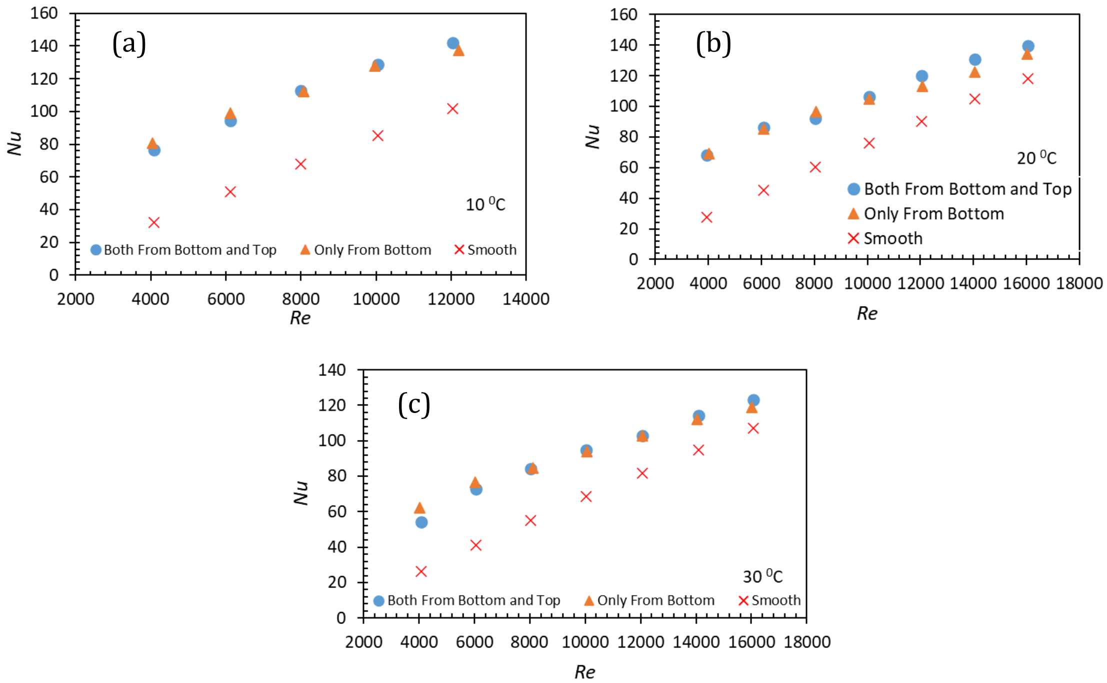


Figure 5.6 Comparison of Nusselt numbers for different ways of heat supply at different temperatures: a) 10 °C, b) 20 °C, c) 30 °C

5.4 Heat sink Pressure Drops

It is well known that when trying to increase heat transfer rate by increasing the heat transfer area through extended surfaces, the pressure drop tends to increase.

Therefore, the pressure drop values between the inlet and outlet of the heat sink should be considered in its design. In Figure 5.7, the pressure drop values decrease with increasing water temperature, similar to *Nu* and *h*. However, although it is desirable for the heat sink to have high *Nu* and *h* at low temperatures, it is undesirable to have high pressure drop at low temperatures. How this affects the performance evaluation criterion will be examined in the following sections.

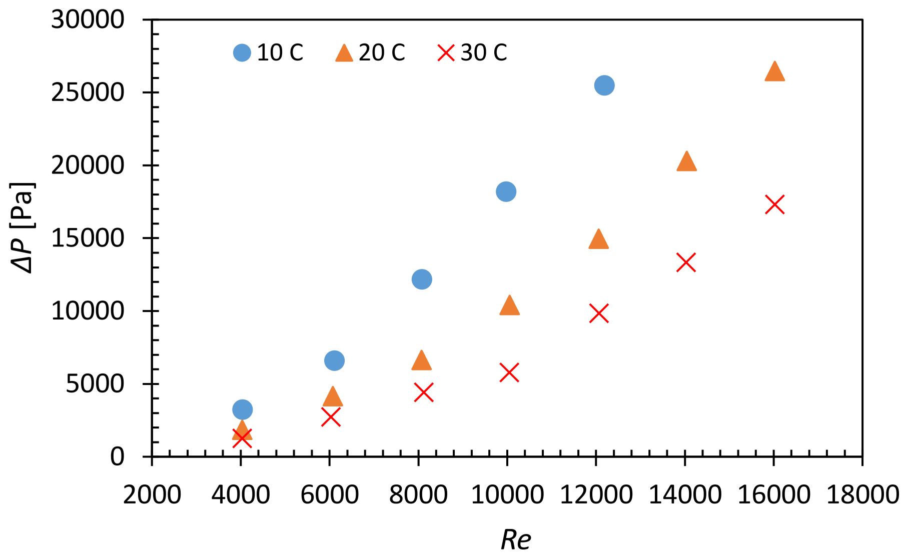


Figure 5.7 Effect of water temperature on pressure drop when heat is supplied only from the bottom

The pressure drop does not change with the way of heat supply and it can be seen clearly in Figure 5.8.

5.5 Friction Coefficient

In order to compare the heat sinks more accurately, friction coefficient values are given and these values are used in the calculation of the performance evaluation criterion (FOM).

In Figure 5.9, the coefficient of friction does not change with water temperature and this is consistent with the experiments of Wang et al. [7], because in these experiments, after the transition region (turbulence region), the friction coefficient does not change with temperature.

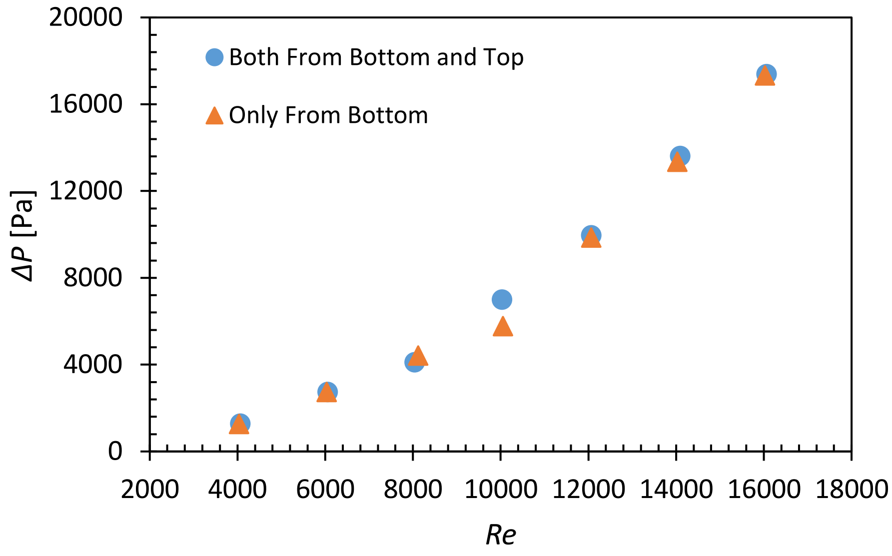


Figure 5.8 Pressure drop in relation to *Re* and the way of heat supply at 30 °C

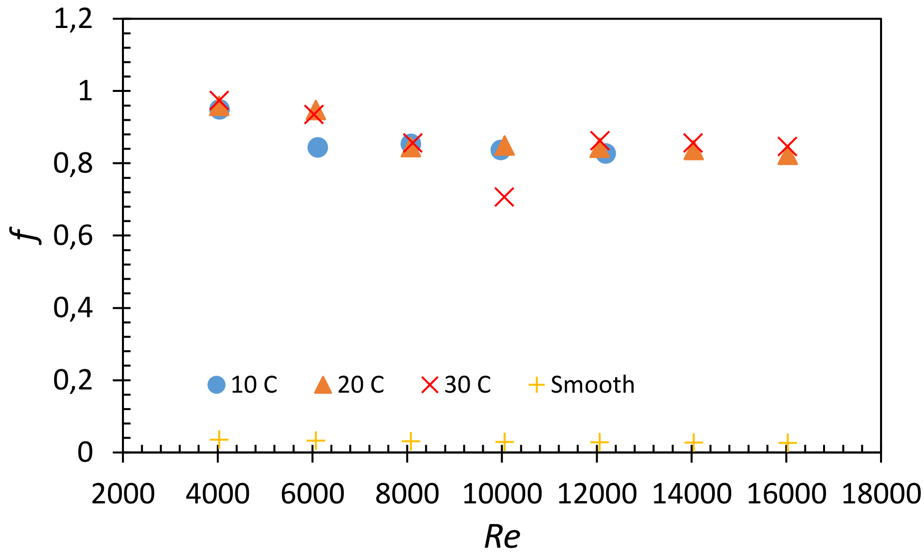


Figure 5.9 Variation of *f* with water temperature and *Re*, including friction coefficient of smooth rectangular channel

As with the pressure drop values, the way of heat supply has no effect on *f*, as can be seen in Figure 5.10.

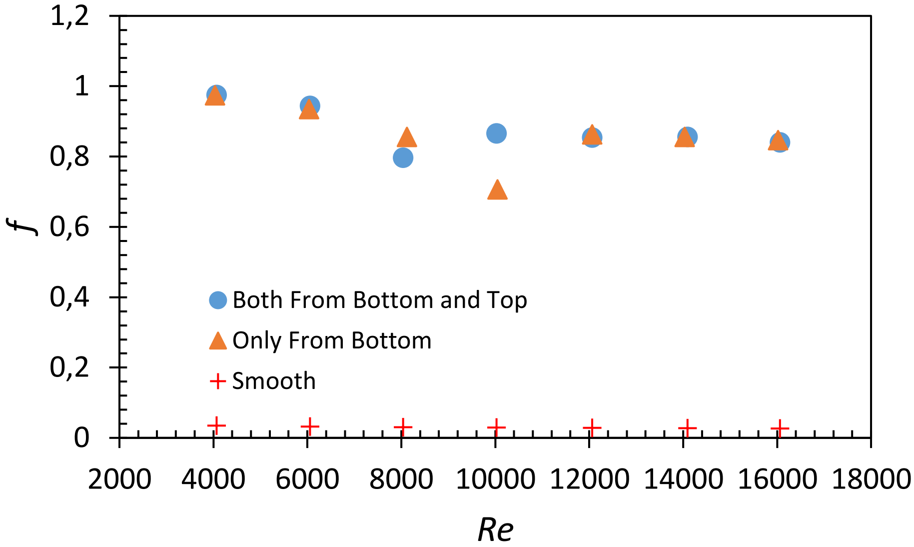


Figure 5.10 Friction coefficient variations with the way of heat supply and *Re*, including friction coefficient of smooth rectangular channel

5.6 Thermal Performance of the Heat sink by Considering the Total Thermal Resistances

Low thermal resistance means that heat can easily pass through the material. Therefore, low thermal resistance increases thermal performance. Firstly, the "both from top and bottom" case is examined. The top of the heat sink has a higher thermal resistance than the bottom, as can be seen in Figure 5.11. This is because the bottom part has much more heat transfer surface, which leads to an increase in convection. It should be noted that thermal resistance is inversely related the convection heat transfer. So, the thermal resistance decreases as the water temperature decreases, this means that the heat transfer rate increases, the same increasing trend can be seen in the *Nu* and *h* figures. In conclusion, the heat transfer performance is better when the water temperature is reduced.

Since there is only one heat supply surface for the "only from bottom" case, there is only one thermal resistance and Figure 5.12 shows only the bottom side of the thermal resistance. The thermal resistance decreases as the water temperature increases. Also in Figure 5.12, the thermal resistance of the bottom surface is lower than the thermal resistance of the "both from top and bottom" case.

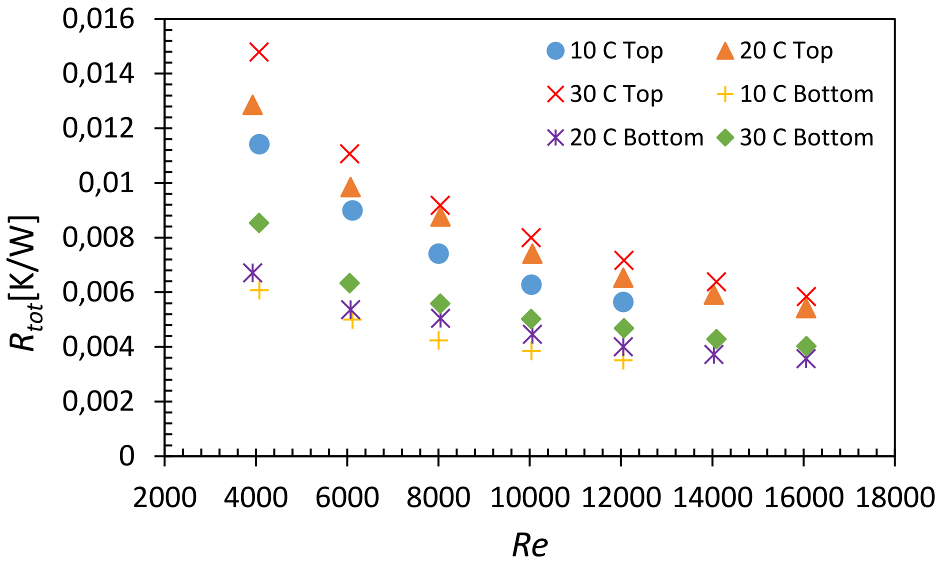


Figure 5.11 The variation of the thermal resistance with the water temperature and *Re* for the "both from top and bottom" case

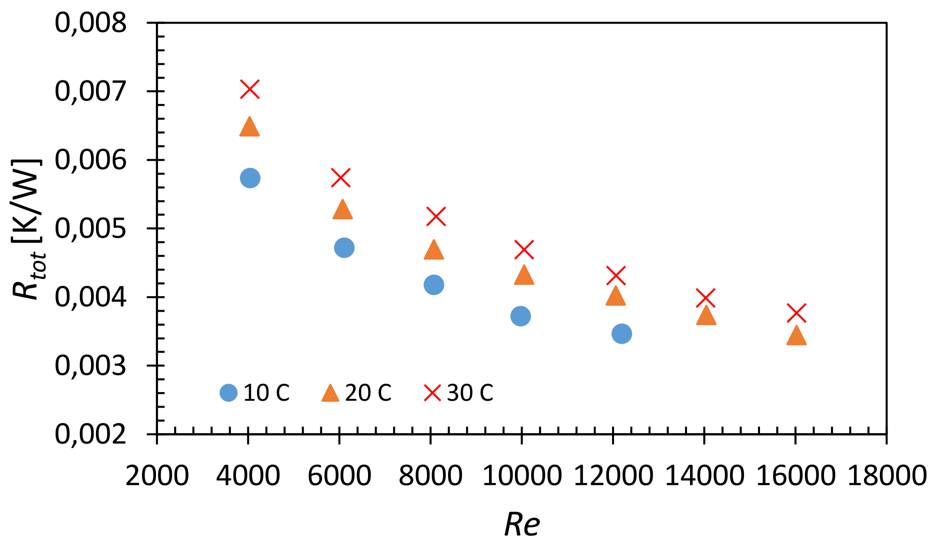


Figure 5.12 The variation of the thermal resistance with the water temperature and *Re* for the "only from bottom" case

5.7 Performance Evaluation Criterion (FOM)

Figure of Merit (FOM) is used to understand the hydrothermal performance of the heat sink and how the changes affect the overall performance. FOM includes both heat transfer and pressure drop values, which are 2 important factors for the heat sinks. When we make a comparison of the water temperature on the overall performance of the heat sink, as can be seen in Figure 5.13, the water temperature has almost no effect on the overall performance of it. However, only at 10 °C does the FOM increase slightly. This is because both *Nu* and *ΔP* increase with decreasing water temperature.

When Figure 5.14 is examined, it is seen that the way of heat supply does not affect the FOM starting from the 6000 *Re*, but only bottom heat supply increases the FOM at 4000 *Re*.

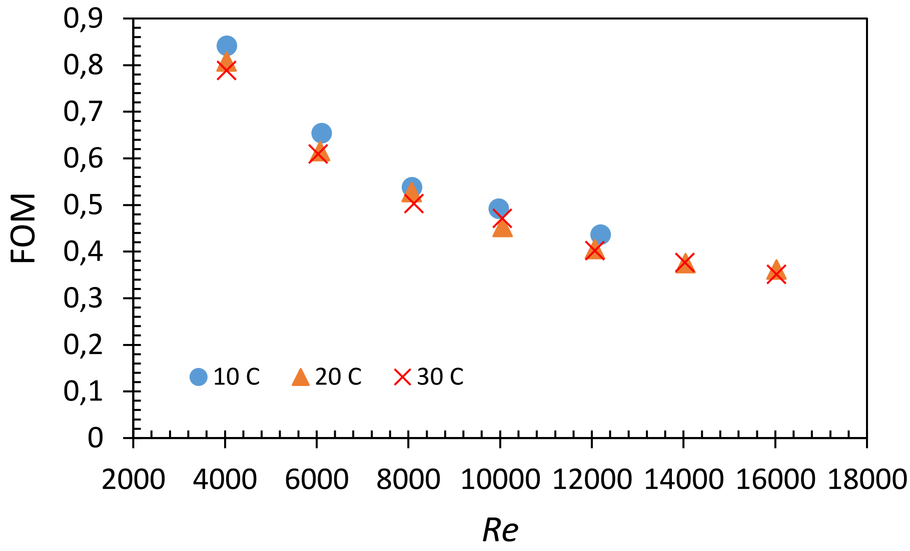


Figure 5.13 When the heat is supplied only from the bottom, change in FOM with respect to *Re* and water temperature

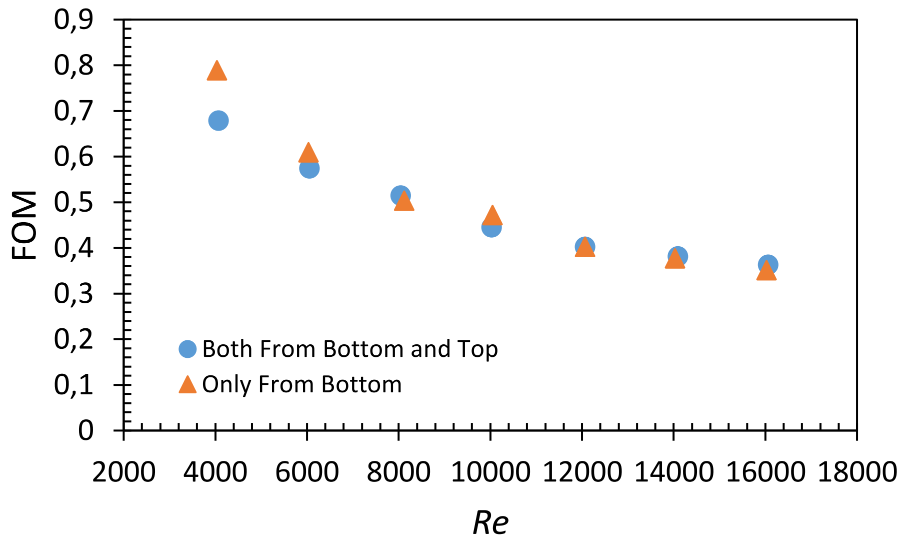


Figure 5.14 Variation of FOM with respect to the way of heat supply and *Re* at 30 °C

6
CONCLUSIONS AND RECOMMENDATIONS

In this thesis, a miniature heat sink with staggered rectangular pin-fins is experimentally examined for use in battery cooling. Before the heat sink, the length of the rectangular channel is deliberately kept long enough to provide the heat sink with a fully developed flow. Inside the heat sink, a new feature in the literature for water cooled heat sinks, is the fin tip clearance between the rectangular pin fins and the top of the heat sink. The experiments are conducted at high Reynolds numbers.

Experiments are carried out to determine the best conditions for different water temperatures, ways of heat supply and Reynolds numbers. After that; temperature differences, convection heat transfer coefficient, Nusselt numbers, pressure drops, friction coefficients, thermal resistances and FOMs are shown in the figures and commented. Here are the conclusions and recommendations of this thesis:

- The temperature difference is likely to be low on the finned surface of the heat sink but, high on the non-finned surface of the heat sink at the laminar and transitional flows. However, especially from the beginning of the 6000 *Re* (at the turbulent flow), the temperature difference on the non-finned surface begins to be lower than that on the finned surface of the heat sink, and at higher *Re* the temperature difference on the non-finned surface becomes almost zero. Therefore, in laminar and transitional flow, it is good to use fin structures on both sides of the heat sinks, but in turbulent flow (especially at high *Re*), it is good to use fin tip clearance to reduce the surface temperature difference. The fin tip clearance is likely to create more turbulence on the surface of the heat sink at high *Re*, causing the temperature difference to deteriorate. To reduce the surface temperature difference at high *Re* for the heat sinks used for battery cooling, the fin tip clearance can be a good way to be used and this heat sink includes only one fin tip clearance. In future studies, the heat sink with fin tip clearance on both sides of the heat sink can be used and both the temperature difference and the performance of the heat sink can be compared with the finned heat sink.

There is probably no need for the heat sink to include the fin tip clearance in any case. Structures that increase the turbulence near the surface of the heat sink are probably sufficient to reduce the surface temperature differences. This could be a very important output for the battery thermal management systems.

- Both *h* and *Nu* increase with increasing *Re,* as expected, and also with decreasing water temperature, their values increase, so it is good to use low fluid temperature to improve the heat transfer rate and it can be used in batteries to reduce their temperatures in shorter times. The heat supplied ways have no effect on *Nu* and *h* when their curves are compared.
- The heat supplied ways have no effect on the pressure drop values. The pressure drop values increase with increasing *Re*, as expected. The decrease of the inlet water temperature has a negative effect on the pressure drop values. So, if it is required to reduce the parasitic load (energy supplied to cool the batteries), low water temperatures and low flow rates can be used. To keep the batteries in the desired temperature range, high water temperatures may be used for low parasitic load after the battery temperatures have been reduced to the desired temperature range.
- The friction coefficient does not change with the temperature of the water and the heat supplied ways. As the flow rate (also *Re*) increases, the friction coefficient has almost no change, it has stabilized as expected. However, the friction coefficient values are higher than our expectations.
- The total thermal resistance decreases as the flow rates increase. The total thermal resistance of the bottom side of the heat sink (finned surface) is less than the upper side of the heat sink. As the water temperature decreases, the total thermal resistances also decrease. Heating from the bottom only reduces the total thermal resistances compared to heating from both the top and bottom. Therefore, increasing the flow rate, placing fins to the heat sink, reducing the water temperature and changing the heat supply way will result in an increase in the heat transfer rate. It has been said that using the fin tip clearance can reduce the temperature difference of the heat sink in the turbulent flow, but it is also seen that this can cause the heat transfer rate to decrease, so it should be optimized or new shapes should be found to both

reduce the water temperature difference and not reduce the heat transfer rate.

- When looking the overall performance of the heat sink by looking at the FOM values: 20 and 30 °C water temperatures have the same values, but 10 °C has slightly higher FOM values than 20 and 30 °C, so using the water temperature at 10 °C can be slightly more logical. The heat supplied way does not affect the overall performance, but only at 4000 *Re* does the 'only from bottom' case has a higher overall performance than the 'both from bottom and top' case and this point that it can be beneficial to use 'only from bottom' heat supplied way in laminar and transitional flows. Also, using lower *Re* values in turbulent flow improves the overall performance and it means that the overall performance of laminar and transitional flows should be examined for the chance of further improvement of the overall performance. It should be noted that many performance evaluation criteria are used in the literature and which one is the best to evaluate the performance of the heat sink should be determined and it can be determined by comparing their entropy generation rates and examining their second law values in thermodynamics.

The aim of this thesis is to speed up the development of heat sinks and heat exchangers. The fin tip clearance is tried for a new thing in the literature and find that the fin tip clearance can reduce the temperature difference through the plate of the heat sink in the turbulent flow and this is very important thing for the battery cooling, but the fin tip clearance can also decrease the heat transfer rate, so it should be optimized if it wants to be used, or new type of fin which increases the turbulence around the plates of the heat sink but not decrease the heat transfer rate should be found. It was also found that reducing the water temperature would immediately reduce the battery temperatures. The difference in heat supplied way has almost no effect in turbulent flow.

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PUBLICATIONS FROM THE THESIS

Conference Papers

1. ÜRKMEZ, Ş. and GEMİCİ, Z., “Experimental Investigation of Heat Transfer and Pressure Drop Characteristics of a Rectangular Narrow Channel Type Heat-Exchanger with Rectangular Staggered Fins,” in Global Summit on Advanced Materials & Sustainable Energy (G-AMSE22), Y. AKINAY, Ed., Van, Oct. 2022, pp. 101–111.

Projects

1. “Batarya Termal Yönetim Sistemlerinde Kullanılan Soğutma Amaçlı Minyatür Pin Kanatlı Kanallarda Tek Fazlı Isı Transferinin Deneysel Olarak İncelenmesi”, Yildiz Technical University Scientific Research Projects Coordination Department, Grant No: FYL-2022-4879