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EFFECT OF GEOMETRICAL PARAMETERS ON NANOFLUID BASED NATURAL CIRCULATION LOOP PERFORMANCE

DEPARTMENT OF GRADUATE SCHOOL OF NATURAL AND APPLIED SCIENCES

M.Sc. THESIS

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JUNE 2019

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İZMİR KATİP ÇELEBİ ÜNİVERSİTESİ
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GEOMETRİK PARAMETRELERİN NANOAKIŞKANLI DOĞAL TAŞINIM DÖNGÜSÜ PERFORMANSINA ETKİSİ

YÜKSEK LİSANS TEZİ

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MOHMMAD ALABOUD, a **M.Sc.** student of **IKCU Graduate School Of Natural And Applied Sciences**, successfully defended the thesis entitled "**EFFECT OF GEOMETRICAL PARAMETERS ON NANOFLUID BASED NATURAL CIRCULATION LOOP PERFORMANCE**", which he prepared after fulfilling the requirements specified in the associated legislations, before the jury whose signatures are below.

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FOREWORD

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The goal of the thesis is studing the SPNCmLs thermo-hydraulic performance considering the effect of geometrical parameters (pipe diameter and aspect ratio), and nanofluids as the working fluid. This study provides choosing the appropriate geometrical parameters and working fluids for SPNCmLs in desired heat transfer application. I would like to thank to Nur Çobanoğlu for her supports and her big role to make it done. Finally, I would like to thank my parents, Issam ALABOUD and Ftoun OBAJI, brothers, and my sister supporting me and being a source of motivation and inspiration in my life.

January 2019 Mohmmad ALABOUD

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ABBREVIATION

A:	Cross sectional area (m^2)
AR:	Aspect ratio
Al_2O_3 :	Aluminium oxide
***b*:**	Constant of $f = p/Re^b$
c:	cooler
c_p:	Specific heat (J/K)
c:	Corrected
D:	Hydraulic diameter (mm)
DIW:	De-ionized water
e:	Effective
f:	Fluid
G:	Gravitational acceleration (m/s^2)
Gr_m:	Grashof number
H:	Height (mm)
k :	Thermal conductivity of the fluid (W/m.K)
L:	Length (mm)
l:	Liquid
m:	Mass flow rate (kg/s)
N_G:	Effective lost coefficient
NCL:	Natural circulation loop
Nu:	Nusselt number
p :	Particle
Pr :	Prandtl number
***p*:**	Constant of $f = p/\mathrm{Re}^b$
Re_{ss} :	Steady state Reynolds number
Ra :	Rayleigh number
r:	Reference
St :	Stanton number
SPNCL:	Single-phase natural circulation loop
SPNCmL:	Single-phase natural circulation mini loop
ss:	Steady state
T:	Temperature (°C)
TPNCL:	Two-phase natural circulation loop
T_{max} :	Maximum temperature
ΔT_{heater} :	Temperature difference
U_{in}:	Overall heat transfer coefficient ($W/m^2.k$)
V:	Velocity (m/s)
vol:	Volume
wt:	Weight

θ:	Inclination angle (°)
ε:	Effectiveness
ρ_e **:**	Effective density (kg/m^3)
β_e **:**	Effective thermal expansivity (1/K)
ϕ **:**	Volumetric concentration
ξ **:**	Local heat transfer loss

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EFFECT OF GEOMETRICAL PARAMETERS ON NANOFLUID BASED NATURAL CIRCULATION LOOP PERFORMANCE

ABSTRACT

This study aims to investigate the effect of pipe diameter and aspect ratio (AR) on the thermal performance of nanofluid-based on single-phase natural circulation mini loop (SPNCmLs).

In the first part of the study, literature review about affecting parameters on natural circulation loops and nanofluid based SPNCLs has been presented.

In the second part, developed 3D model for steady state conditions has been defined. Numerical results obtained by developed model, which is based on experimental study in the literature, were compared with the experimental results of the same study. In this study, deionized water based nanofluid which has different volumetric concentrations of Al_2O_3 nanoparticles was used. The thermophysical properties were taken from experimental study in the literature. Moreover, in this part, the performance parameters of SPNCLs have been explained.

As a result, the heating power, concentration of nanofluid, pipe diameter and AR affect the performance of SPNCmLs. The performance of SPNCmL was evaluated in terms of temperatures, mass flow rate and heat transfer coefficient as dimensional parameters and effectiveness factor as non-dimensional parameter. For lower pipe diameters and AR values, system has better performance. Although, using nanofluid increased temperatures and $\boldsymbol{\varepsilon}$, it decreased mass flow rate and overall heat transfer coefficient. It is needed to define the performance of SPNCmL by non-dimensional parameters. Moreover, by this study favourable geometrical parameters and working fluids can be chosen for highly efficient SPNCmL in desired heat transfer application.

GEOMETRİK PARAMETRELERİN NANOAKIŞKANLI DOĞAL TAŞINIM DÖNGÜSÜ PERFORMANSINA ETKİSİ

ÖZET

Bu çalışmada, boru çapı ve en/boy oranının nanoakışkan bazlı tek fazlı doğal taşınımlı mini döngünün (TFDTmD) ısıl performansı üzerine etkisi amaçlanmıştır.

Çalışmanın ilk bölümünde, doğal taşınım döngülerine etki eden faktörler ve nanoakışkan bazlı TFDTD ile ilgili literatür araştırması yapılmıştır.

İkinci bölümünde ise kararlı hal için oluşturulan 3 boyutlu model hakkında bilgiler verilmiştir. Literatürde yer alan deneysel çalışma baz alınarak oluşturulan modelden elde edilen sayısal sonuçlar deneysel çalışmanın sonuçları ile karşılaştırılmıştır. Çalışmada farklı hacimsel konsantrasyonda Al_2O_3 nanoparçacıklarına sahip deiyonize su bazlı nanoakışkan kullanılmıştır. Termofiziksel özellikleri literatürdeki deneysel çalışmadan alınmıştır. Ek olarak, bu bölümde, TFDTD'nin performans parametreleri açıklanmıştır.

Sonuç olarak, ısıtıcı gücü, nanoakışkan konsantrasyonu, boru çapı ve en/boy oranı TFDTmD'nin performansını etkilemektedir. TFDTmD performansı boyutlu parametre olan sıcaklıklar, kütlesel debi ve ısı transferi katsayısı dışında boyutsuz parametre olan etkinlik katsayısı açısından değerlendirilmiştir. Düşük çaplarda ve en/boy oranında sistem daha iyi performansa sahiptir. Nanoakışkan kullanımı sıcaklık değerleri ve etkinlik katsayısını arttırsa da kütlesel debi ve ortalama ısı transferi katsayısını azaltmaktadır. TFDTmD performansının boyutsuz sayılar ile genelleştirilerek değerlendirilmeye ihtiyacı vardır. Ayrıca bu çalışma ile yüksek verimli TFDTmD kullanılacak ısı transferi uygulamaları için uygun geometrik parametreler ve çalışma akışkanları seçilebilir.

1. INTRODUCTION

Development of the technology enables the increase in heat transfer efficiency and performance of the devices, but heat transfer problems still exist from macro to nanoscales. Although, force convection systems have been prefered more than passive cooling systems in macroscale devices, as the size of devices is getting smaller and more compact, passive cooling systems are getting attraction due to the diffuculty of using additional parts. Natural circulation loops (NLCs) or thermosyphons are the most widely known passive systems. The system works on transfering the heat from a hot medium to a cold medium by natural convection. Density gradient based buoyancy forces are driving mechanism of NCLs instead of external mechanical forces. There are two types of NCLs: single-phase natural circulation loops (SPNCLs) and two-phase natural circulation loops (TPNCLs). In SPNCLs, density gradient is caused by temperature gradient whereas in TPNCLs the main reason behind density gradient is the phase change. SPNCLs are simpler, easier to control, safer and more reliable compared to TPNCLs. With these advantages, they have been used in cooling of nuclear reactors, solar water heaters and electronics cooling applications.

In the SPNCL to provide stable flow and enhanced heat transfer many studies have been carried on the effect of the geometry and the fluids materials. The geometry as primary factor is studied in terms of pipe diameter, loop shape, inclination angle, heater and cooler orientation, and aspect ratio (AR) of the loop to improve stability and performance [1]. Vijayan et al. [2] studied the effect of loop diameter (6, 11, 23.2, and 26.9 mm) on the stability of SPNCL and TPNCL by considering the flow rate and steady state Reynolds number changes. They observed instability only for two large diameter loops and found that the instability threshold (the limit where the instability begins) is found to decrease with increasing loop diameter. Thus, it is concluded that small diameter loops are more stable than large diameter loops. As a one of the geometrical characteristics, miniaturizing has also been investigated.

Misale et al. [3] proposed mini SPNCL (SPNCmL) for the first time by and they reported that SPNCmLs have higher stability compared to SPNCLs. Dimensions of mini loops was determined as 4 mm of pipe diameter, 100 mm of horizontal legs, 180 mm of width and 264 mm of height. They studied various powers (2.5 W, 5 W, 7.5 W, 10 W, 15 W and 25 W) and inclination angles (0°, 30°, 75°). For each experimental run, the thermal-hydraulic behavior of water as a working fluid was stable. When the loop was inclined 75°, temperature was affected, and instabilities occurred during the transient case. The most effective thermal performance obtained when the inclination angle was 0° and power equaled to 25 W.

Garibaldi and Misale [4] work experimentally with the distilled water and FC-43 as working fluids to study the thermal performance of mini loops, changing in both heat transfer rate to the fluid and mini-loop inclination. On the other hand, the stability of thermo-hydraulic behavior for both fluids and all combinations analysed. They found heat flux density of about 40 kW/m^2 with distilled water was the best thermal performance of the mini-loop. They also compared the mass flow rate and all the data also showed a very good agreement with the Vijayan's correlation for mass flow rate [2]. Beside them, AR as another geometrical parameter influences the thermo-hydraulic performance of the NCL. Doganay et al. [5] presented that **ε** increases up to 20% with the decrease of AR from 1.38 to 0.4 in SPNCmLs. Garibaldi [6] also studied with SPNCmLs with AR (ML1=0.84 and ML2=1.47). With the increase in loop height (ML2), higher fluid velocity was expected but not observed. Because enhancement in shear stress compensated the buoyancy forces in higher loops. Bieliński and Mikielewicz [7] reported that flow rate increases with the higher AR for SPNCL (with orientation of horizontal heater vertical cooler) at constant total pipe length.

As mentioned above, variable working fluid is another option to increase heat transfer performance. Nanofluids, as new type of heat transfer fluids, have been getting attraction since 1995 when Choi [8] introduced the term of "nanofluid" for the first time. It can be defined as a suspension of nanosized particles (metals, metal oxides, carbon etc.) in a conventional base fluid (water, oil, ethylene glycol). Both experimental and numerical studies on their effect on stability and thermo-hydraulic performance of SPNCLs have been carried out for 10 years. In the first study of nanofluid based SPNCL, Nayak et al. [9] used water-based Al_2O_3 nanofluid (0.3-2 wt% and particle size 40-80 nm) in rectangular SPNCL. Nanofluids even at low concentrations could suppress the instabilities which occurs by using water. Steady state mass flow rate (20-35%) and pressure drop (28-44%) increased with particle concentration. Nayak et al. [10] have used water-based Al_2O_3 nanofluid (1 wt%) and similar to previous study they obtained stable flow regime for all cases in which nanofluid used. But when water was used as a working fluid, they have observed both flow regime instability and phase-change instability. Thomas and Sobhan [10] used water-based Al_2O_3 and CuO nanofluids as a working fluid in their the study. Increases in steady state mass flow rates of nanofluids were calculated as 12% and 14% for 0.075 vol% of alumina-water and 0.05 vol% of CuO-water nanofluid, respectively. The dispersion stability is reported as important factor for choosing working fluid. They claimed that the reasons behind higher heat transfer properties of Al_2O_3 nanofluid could be dispersion stability.However, increase in concentration and heat input, and decrease in tilt angle results in higher Ra number and average heat transfer coefficient. In addition to enhanced stability of the SPNCmLs, nanofluids have been used to obtain higher thermo-hydraulic performance of SPNCmL.

Misale et al. [11] have used water-based Al_2O_3 nanofluid (0.5 & 3 vol.%, 45 nm) in SPNCmL at different inclination angles (0°to 75°) for the first time. Miniaturization resulted in stable flow regime for both water and nanofluids. For 75° inclination angle the performance of SPNCmL improved slightly but nanofluids have comparable performance with water. However experimental results are in good agreement with Vijayan's correlation [2]. Turgut and Doganay [12] proposed the effectiveness factor (ε) which is based on temperature differences for evaluation of thermal performance. They used Al_2O_3 nanofluid (1, 2 & 3 vol.%, 10 & 30 nm) in SPNCmL at 0° inclination and between 10-50 W of power inputs. Addition of nanoparticles into basefluid increases the ε factor for all cases. Increase in power input resulted in reduction of **ε** factor. They reported that concentration had considerable effect on ε factor for larger particles. Karadeniz et al. [13] proposed a numerical 3D model for SPNCmL which is based on Turgut and Doganay [12] setup. They used water based Al_2O_3 nanofluid as a working fluid. Filler content was determined as 1, 2 and 3% for nanofluids. They studied the temperature and **ε** factor as performance parameters. T_{max} in this study had same tendency with the experimental results of [12] but values were over-predicted due to the heat loss to surrounding medium in experimental result. Also because of thermal conductivity of 3%, over prediction increased. Ho et al. [14] used mini-channel heat sink and heat source in a SPNCmL and showed that higher concentration increases the average Nu number for both heater and cooler sections. Nu number decreases by increasing temperature in heater sections but in cooler section it increases. For constant Ra, addition of nanoparticles decrease Re. Doganay and Turgut [15] evaluated **ε** factor in terms of inclination angle (0°, 30°, 60° and 75°), heating power (10, 30, 50 W) and heat sink temperature (10°C and 20°C). They used Al_2O_3 nanofluid (1, 2 & 3 vol.%, 10 & 30 nm). Increase in inclination angle and concentration increased **ε** factor at all powers. Karadeniz et al. [18] proposed numerical study on the experimental results of Doganay and Turgut [17] by using their previous model [15]. Numerical results were good agreement with the experimental data except 50 W and 75° inclination angle. The possible reason behind this unexpected behaviour was reported as the phase change in the heater section.

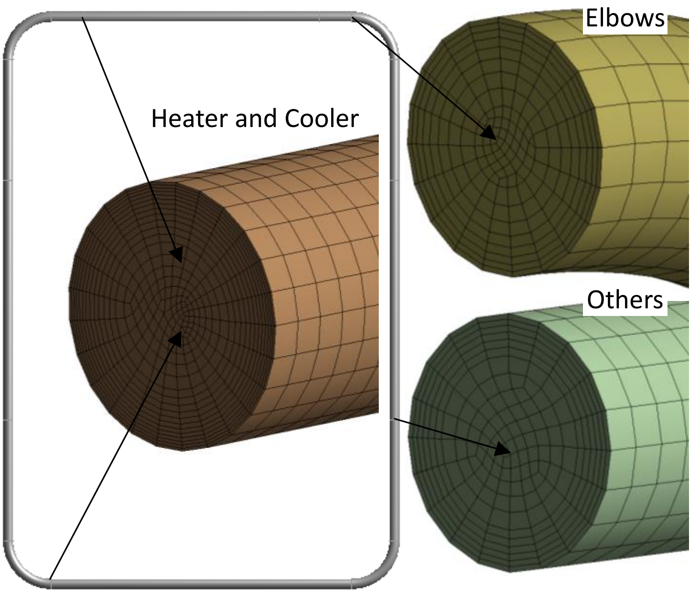
Koca et al. [16] studied with water based Ag nanofluid (0.25-1wt%.) in the same setup with [12; 15]. Using lower concentrations of nanofluids and change in inclination angle did not have significant effect on thermal stability. Higher concentrations of Ag-nanofluid have great effect on **ε** compared to Al_2O_3 in same SPNCmL setup. On the other hand, Mohan et al. [17] used mineral oil based- hexagonal boron nitride (hBN) nanofluid (0.02 & 0.08 wt%.) in SPNCmL which have AR of 0.77. Increase in power resulted in higher ΔT_{heater} by 2-7.4% compared to water. By using nanofluid (0.08 wt.%), flow rate increased 4% and maximum Nu number enhancement was 5%.

In perusing the available literature, it seems that there is a gap in numerical nanofluid based SPNCmL studies focused on the AR and pipe diameters. For the first time, this 3D steady numerical study presents how both AR and pipe diameter affect to thermo-hydraulic performance of the Al_2O_3-DIW nanofluid based SPNCmL. During the study, the total pipe length kept constant and the AR altered between 0.4 and 1.38 And also, AR values of 1.6, 1.8 and 2 which have different total pipe length has been evaluated. Pipe diameters varied between 3 and 6 mm at a constant AR. The analyses were conducted with DIW and Al_2O_3-DIW nanofluids with 1, 2, 3 % volumetric particle concentrations.

2. MATERIALS AND METHODS

Studying parametric effects is easier by using numerical methods compared to experiments to obtain more accurate results. This thesis presents the pre-confirmed numerical study on geometric effects on performance and characteristics of SPNCmL. For investigation of thermo-hydraulic performance of the SPNCmL; temperature difference between hot leg and cold leg, maximum temperature and mass flow rate are evaluated as dimensional parameters. Change in maximum temperature (T_{max}) and temperature difference (ΔT_{heater}) with the heat transfer rate can be useful for comparison of performances of different SPNCLs. Direct measurement of mass flow rate is more difficult compared to temperatures. Therefore, empirical correlations and developed models by numerical studies have been used at different operating conditions. Non-dimensional indicators such as effectiveness, Nusselt number (Nu), modified Grashof number (Gr_m), steady-state Reynolds number (Re_{ss}) and Stanton number (St) are used to evaluate the thermo-hydroulic performance of SPNCLs.

Different meshes were apply for sensibility analysis M1: 745456, M2: 565681, M3: 349296 as shown in Figure 2.1. The reliability of the meshes solution was verified by solving the numerical model with them in three different heater power values (10, 30, and 50 W). All meshes included the elements of the micro-boundary layer as well as the finer grids in the heater and the cooler parts of the heat transfer model and the flow of the boundary layer more precisely. For sensitivity analysis, T_{max} and ΔT_{heater} were considered critical for comparing experimental data and numerical data. All the meshes obtained from numerical simulations. However, the result indicates that there is almost no change in T_{max} and ΔT_{heater} values for the different meshes at different heater power values. Therefore, the mesh dependency test does not continue to smaller mesh numbers. In addition, the convergence criteria for all the provision equations were set to be 10^{-7} for all analyzes.



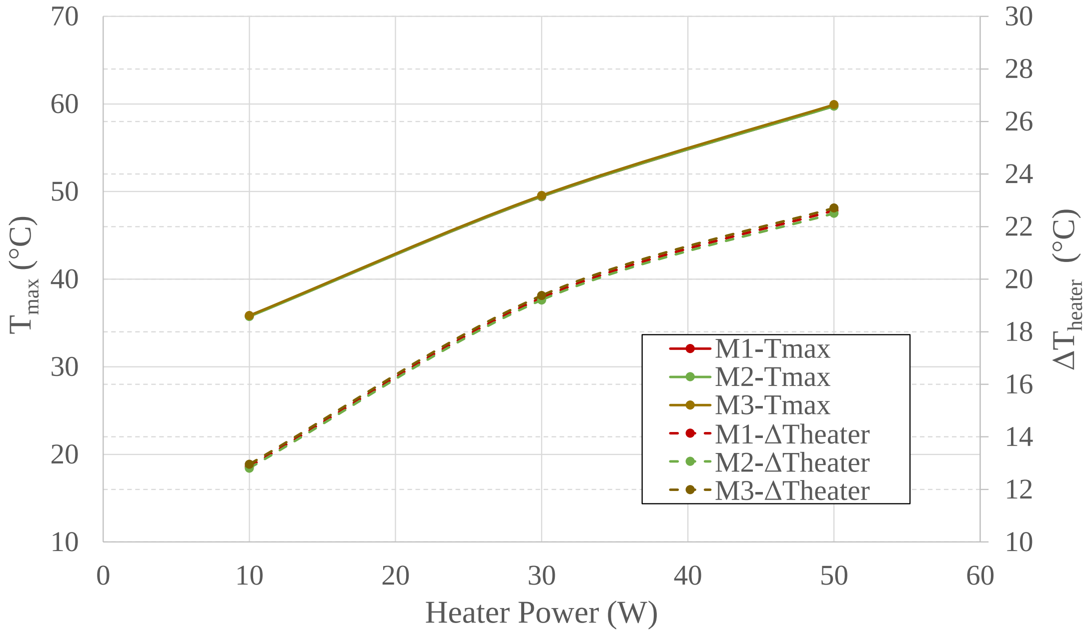


Figure 2.1: Mesh dependency for the heater, cooler section and elbows

2.1 Numerical Study

In this thesis, the effect of pipe diameter and AR on themo-hydraulic performance of SPNCmL has been studied by developing 3D numerical model based on the numerical model [18] and the experimental setup of Doganay and Turgut [15]. combmercial software (ANSYS CFX) was used for analysis and development of the model. Boundary conditions set to constant heat flux at 10, 30, 50 W at heater and constant temperature at cooler part (20°C) similar to experimental study (Figure 2.2).

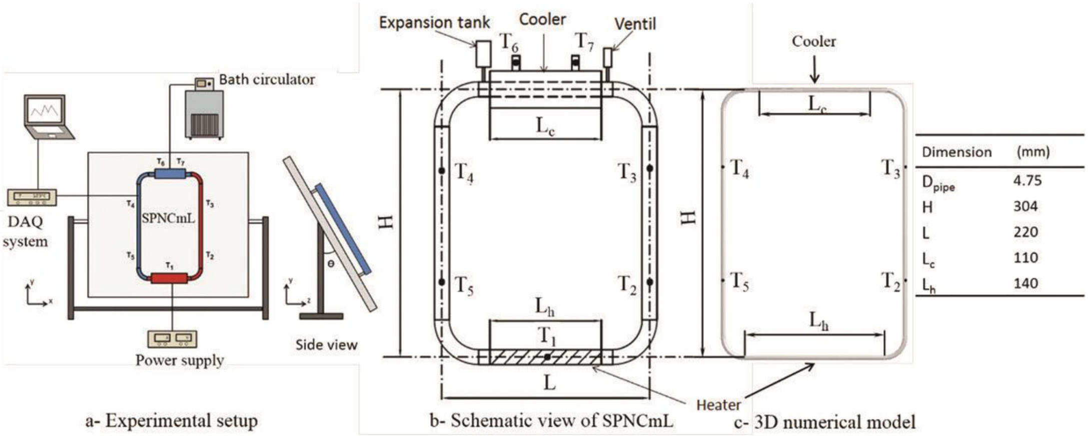


Figure 2.2: Schematic view of experimental setup, corresponding numerical model and dimensions [18]

Table 2.1: The dimension of different AR.

Aspect ratio (H/L)	a [mm]	b [mm]	c [mm]	d [mm]	e [mm]	f [mm]	g [mm]	H [mm]	L [mm]	Total Pipe[mm]*
2	90	180	40	0	140	13	110	440	220	1000
1.8	79	158	40	0	140	13	110	396	220	912
1.6	68	136	40	0	140	13	110	352	220	824
1.38	56	112	40	0	140	13	110	304	220	728
1.2	51.5	103	40	9	140	24	110	286	238	728
1	45.5	91	40	21	140	36	110	262	262	728
0.8	38.2	76.4	40	35.5	140	50.5	110	233	291	728
0.6	29.3	58.4	40	54	140	69	110	197	328	728
0.4	17.4	34.9	40	77	140	92	110	150	374	728

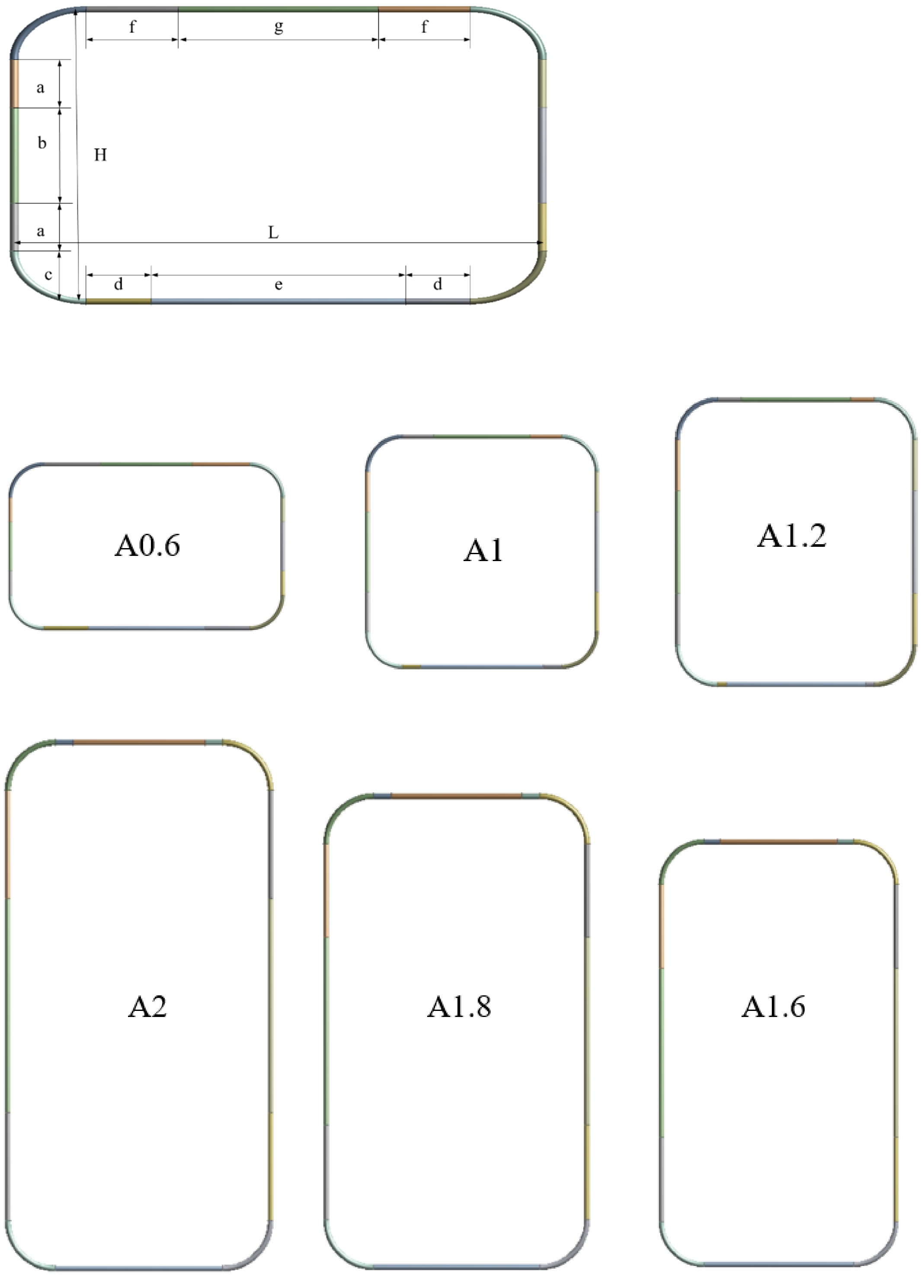


Figure 2.3: AR used in numerical study.

Different AR was used in this study and pipe diameter is 4.75 mm for all AR (Figure 2.3). Total length is held constant for lower AR. As AR is higher than experimental setup, total pipe length is increased.

Viscosity , thermal expansivity, and thermal conductivity as thermophysical proprties are defined as functions of temperature. Effective specific heat was calculated by Eq. 1 [19] and did not change with temperature significantly for both water and nanofluids. The working fluid is water, the tabular values used to define the temperature dependency of properties. For Al_2O_3-DIW nanofluid at different particle concentrations, to determine the effective density (ρ_e) and effective thermal expansivity (β_e).we use the model which is given below (Equation 2.1-2.3).

$$C_e = \frac{\phi_p(\rho C)_p + (1-\phi_p)(\rho C)_f}{\phi_p\rho_p + (1-\phi_p)\rho_f} \tag{2.1}$$

$$\rho_e = (1-\phi_p)\rho_f + \phi_p\rho_p \tag{2.2}$$

$$(\rho\beta)_e = (1-\phi_p)(\rho\beta)_f + \phi_p(\rho\beta)_p \tag{2.3}$$

where ϕ is the volumetric concentration and, f and p denote the fluid and particle, respectively. As thermal conductivity and viscosity in the numerical approach, second order polynomial functions were obtained by using experimental values reported in [20]. Thermophysical properties are presented in the Figure 2.4.

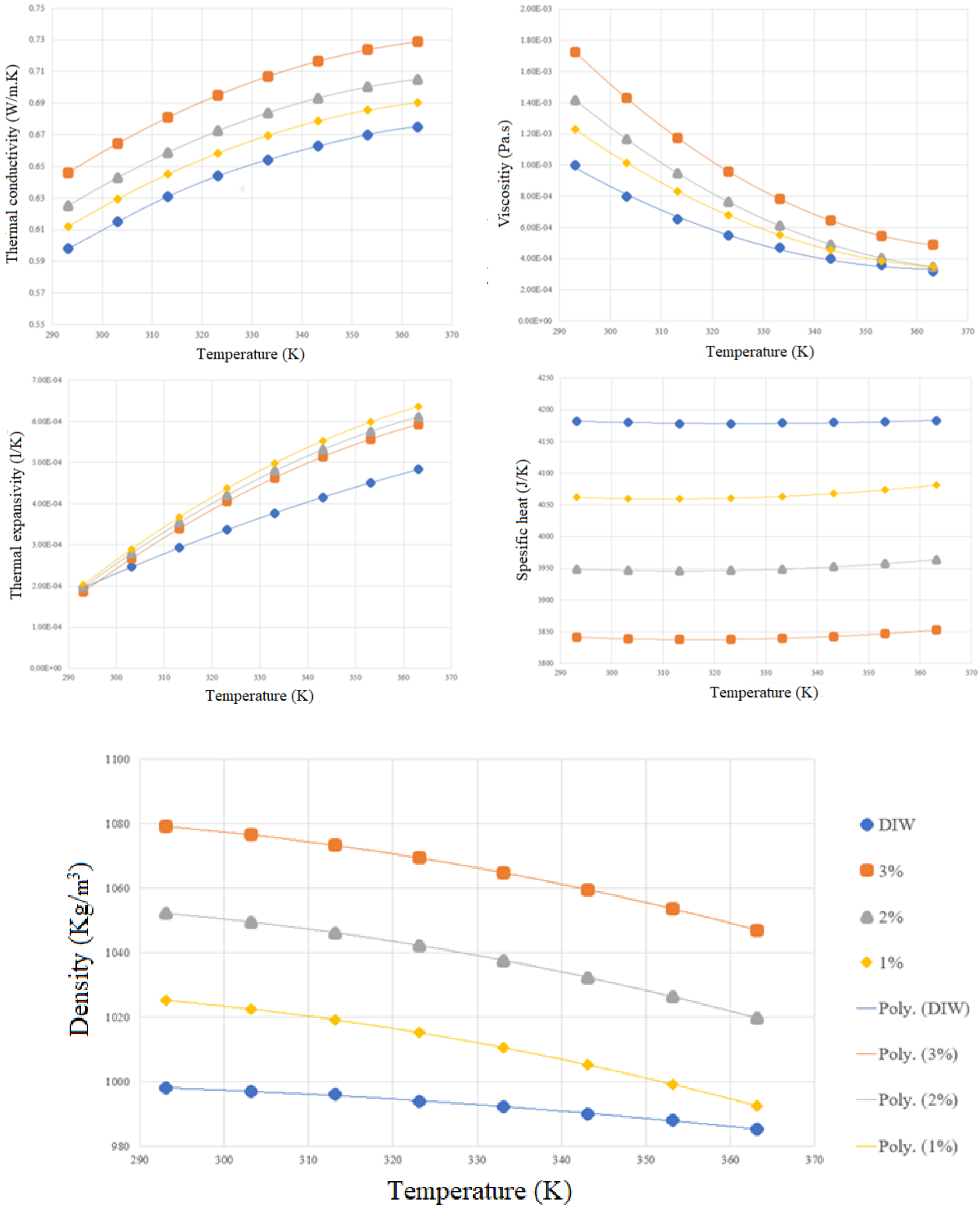


Figure 2.4: Thermophysical properties of working fluids

2.2 Parameter Specification

In this study, different pipe diameter (3 mm, 4 mm, 4.75 mm, 5 mm, 6 mm) on themo-hydraulic performance of SPNCmL 3D numerical model was adopted from [18] the experimental setup of Doganay and Turgut [15]. A commercial software (ANSYS CFX) was used for model. Moreover with this model, effect of AR for constant pipe diameter (4.75 mm) was studied. AR values are varied between 0.4-2.0. Total length is held constant for lower AR. As AR is higher than experimental setup, total pipe length is increased. The heat flux constant at 10, 30, 50 W, as inputs in the heater part whereas in the cooler part, the temperature was defined as 20°C constant in the cooling fluid temperature in the experimental study

2.3 Performance Indicators

ΔT_{heater}, T_{max} and mass flow rate are evaluated as dimensional parameters in accordance with the literature. The main characteristics SPNCmL consideres as maximum temperature and the temperature difference between the outlet and inlet of the heater. The definitive parameter to valuation of SPNCLs performance was the steady state mass flow rate. Besides T_{max} and ΔT_{heater}, $\boldsymbol{\varepsilon}$ can be considered as one of the main characteristics of SPNCmL. Also T_{avg} and ΔT_{avg} (Equation 2.4 and 2.5) are also studied for investigation of thermal performance of the loop [11].

$$\Delta T_{avg} = T_{hot,avg} - T_{cold,avg} \tag{2.4}$$

$$T_{avg} = (T_{hot,avg} + T_{cold,avg})/2 \tag{2.5}$$

Vijayan et. al [2] proposed a theoretical approach to predict mass flow rate. The model is valid for the conditions of natural circulation loops having a uniform diameter, under steady state conditions with an incompressible Boussinesq fluid, and where the local pressure losses can be neglected. The corresponding mass flow rate in the loop is than calculated by

$$\dot{m} = \left[\frac{2\cdot g\cdot D^{b}\cdot \rho^{2}\cdot \beta_{T}\cdot A^{2-b}\cdot Q_{h}\cdot \Delta z}{p\cdot c_{p}\cdot \mu^{b}\cdot N_{G}}\right]^{1/(3-b)} \tag{2.6}$$

Here, N_G is effective loss coefficient and can be calculated as:

$$N_G = \frac{L_{total}}{D} \cdot \frac{\xi}{f} \quad (2.7)$$

where ξ is local heat transfer loss [21].

Non-dimensional parameters are more accurate for comparison of different systems and operating conditions. of the heat exchangers, is a ratio of the actual heat transfer to maximum possible heat transfer. ε, which is a performance indicator.

$$\varepsilon = \frac{T_2 - T_5}{T_2 - T_6} \quad (2.8)$$

To evaluate heat transfer in the SPNCmL heat transfer coefficient is found by using Equation 2.9 where U-in is the overall heat transfer coefficient (W/m^2·K), and k is the thermal conductivity of the fluid (W/m·K) [22]. The overall heat transfer coefficient U_{in} was evaluated on the basis of the logarithmic mean temperature difference and it is referred to the internal area of the cooler. Overall heat transfer coefficient at the cooler was calculated by using the equations below.

$$U_{in} = \frac{Q}{\pi\, D\, L_c\, \Delta T_m} \quad (2.9)$$

$$\Delta T_m = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} \quad (2.10)$$

$$\Delta T_1 = T_{cold,in} - T_c \quad (2.11)$$

$$\Delta T_2 = T_{cold,out} - T_c \quad (2.12)$$

3. RESULTS AND DISCUSSIONS

The results presented the validation of the three-dimensional model by comparing the result of the numerical study and the experimental results in [15]. For different pipe diameters (3 mm, 4 mm, 4.75 mm, 5 mm, 6 mm) and AR (0.4, 0.6, 0.8, 1, 1.2,1.4,1.6,1.8, 2) thermo-hydraulic performance of SPNCmL in terms of performance parameters and temperature contours were discussed. Table 3.1 and Table 3.2 compile symbols of all parameters for pipe diameter and AR, respectively. DIW is represented by blue. Yellow stands for 1% filler content, green is for 2% and orange is for 3%. Moreover, filled symbols are used for 50 W, patterned symbols for 30 W and non-filled symbols for 10 W.

Table 3.1: Symbols for pipe diameters and colors for working fluids.

Symbol	Pipe Diameter [mm]	Color	Working Fluid	Symbol	Power
○	3	●	DIW	○	10 W
△	4	●	1% Alumina NF	⊖	30 W
×	4.75	●	2% Alumina NF	●	50 W
◇	5	●	3% Alumina NF		
□	6				

Table 3.2: Symbols for AR and colors for working fluids.

Symbol	Aspect Ratios	Color	Working Fluid	Symbol	Power
◍	0.4	●	DIW	○	10 W
●	0.6	●	1% Alumina NF	⊖	30 W
▨	0.8	●	2% Alumina NF	●	50 W
▥	1	●	3% Alumina NF		
▩	1.2				
▦	1.38				
▲	1.6				
▲	1.8				
◇	2				

3.1 Validation

The 3D numerical steady-state model was developed by Doganay and Turgut [16] the model based on the experimental setup and results. The experimental study compared with a numerical study result of 4.75 mm pipe diameter in the Figure 3.1. It is seen that T_{max} increases with increasing heater power and increasing volumetric concentration and show good agreement with experimental results of T_{max}. Error of T_{max} varied between ± 20%. The reason could be the heat loss to surrounding medium in the experiments, but in numerical study all the heat generated in the heater part is transferred to the working fluid.

Other reasons could be the measurement techniques and thermophysical properties of working fluid. In the experimental study, the temperature is measured from one point but numerical study allows to determine the temperature from different points. Also, error in the measurement of thermophysical properties could be resulted in the error in the numerical study.

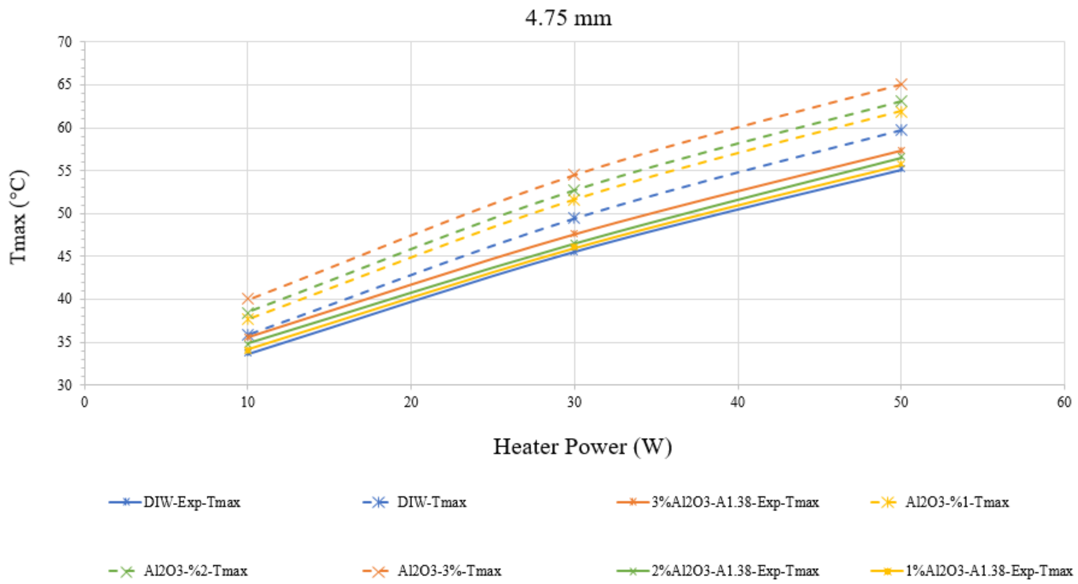


Figure 3.1: Validation of numerical model with experimental values of T_{max} .

The result of the ΔT_{heater} in the numerical study also get compered with the experimental study, generally, the result was similar to the T_{max} the error percentage and the ΔT_{heater} increases with increasing heater power and increasing volumetric concentration. The numerical results are getting approached to experimental results of ΔT_{heater} in the Figure 3.2 at higher powers for nanofluids more than DIW. The values of ε in the numerical model are close to the experimental result and converge more in the power heater 10 W for all the working fluids but difference is increased for 50 W similar to ΔT_{heater} and $T_{max.}$ This may be because of the above mentioned error from the determination of the thermo-physical properties of the nanofluids.

Numerical model's predictions for ΔT_{heater}, T_{max} and ε have consistent deviations from the experimental study for the 1% and 2% volumetric concentrations. Although effectiveness values for 3% volumetric concentration fit well with the experimental study. Besides, the ε increases with the decrease of volumetric concentration and decrement of applied power. In terms of AR, it is clearly seen that ε increases with decreasing AR. Although, the loop length is constant for all AR values which mean the amount of nanofluid in the system is same, the AR influences the ε of the SPNCmL. This reveals the importance of geometry when the SPNCmL designed.

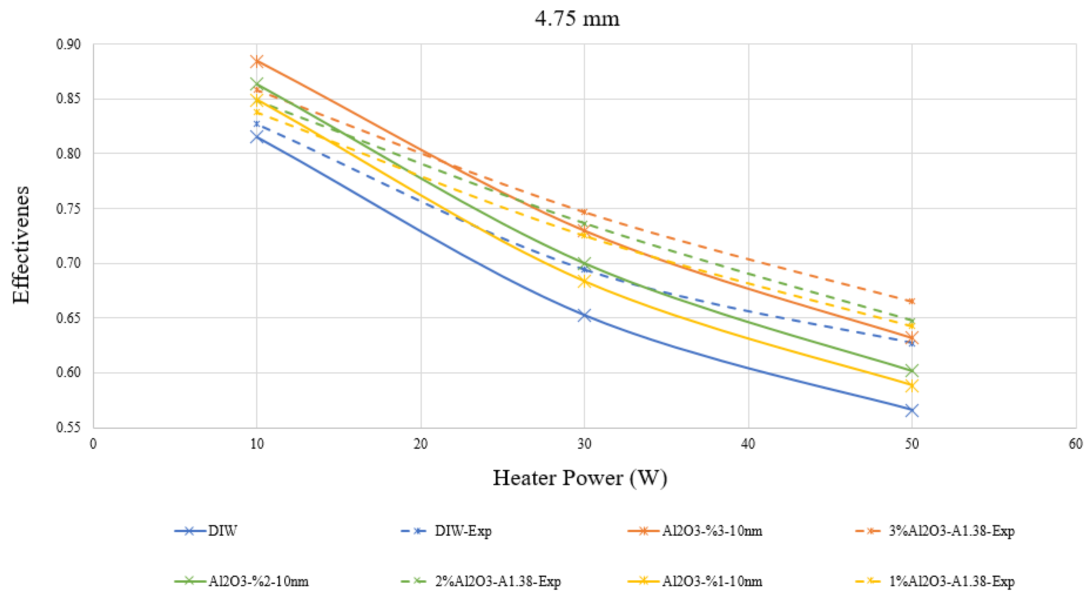


Figure 3.2: Validation of numerical model with experimental values of ε.

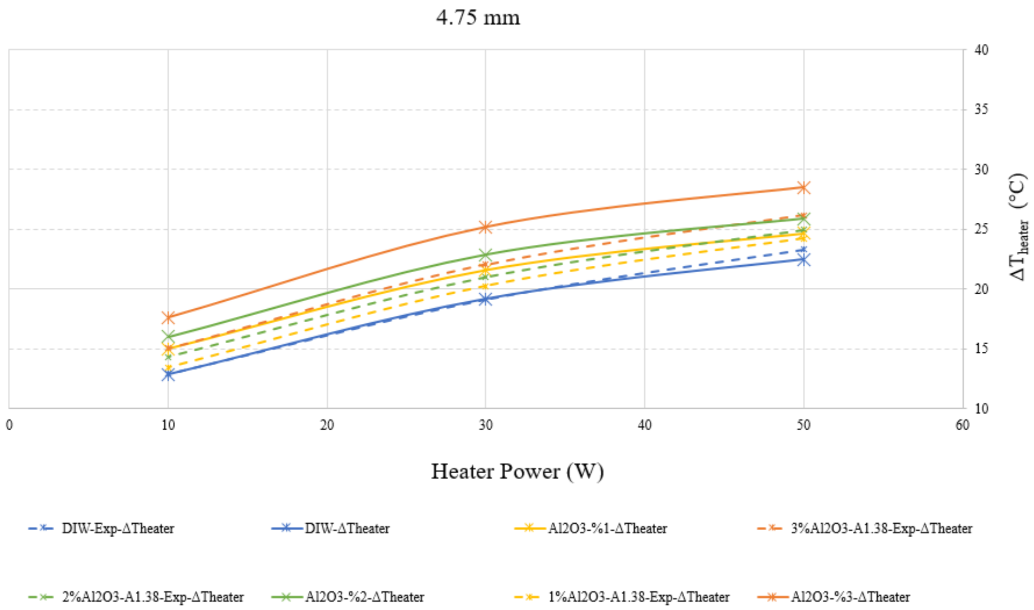


Figure 3.3: Validation of numerical model with experimental values of ΔT_{heater} (°C)

3.2 Performance Parameters

3.2.1 Mass Flow Rate

The mass flow rate of working fluids with all the pipe diameter at all heater powers calculated and correlated with Vijayan's equation at 4.75 mm pipe diameter. The difference between numerical mass flow rate and correlated mass flow rate increases with the increasing of the heater power up to 4 mm. Mass flow rate of nanofluids for all filler contents is smaller compared to DIW. Mass flow rate increases with the pipe diameter in Figure 3.4.

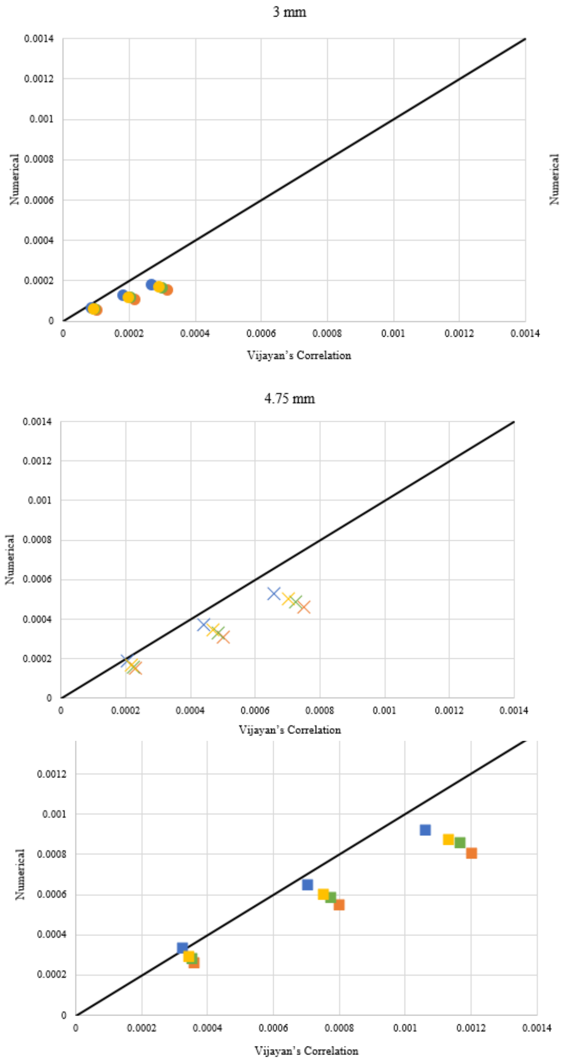


Figure 3.4: Comparison of mass flow rate with Vijayan's correlation for different pipe diameters.

In the literature, it has been reported that mass flow rate of nanofluids is higher than DIW [9, 10, 17, 23, 24]. Thomas et al. [25] presented that increase in concentration increases The possible reason behind this increase behind this behavior as reduction of viscous resistance force and increase in buoyancy forces. There is no significant change in mass flow rate observed for different AR in Figure 3.5. Garibaldi [6] has also reported that higher fluid velocity is expected for highr AR but it was not observed. However, Ho et al. [26] investigated the geometrical parameters affecting the thermal performance of NCL working with phase-change material suspensions. They found that Re number increases with the decrease of AR. Increase of Re number can be explained with increase in mass flow rate.

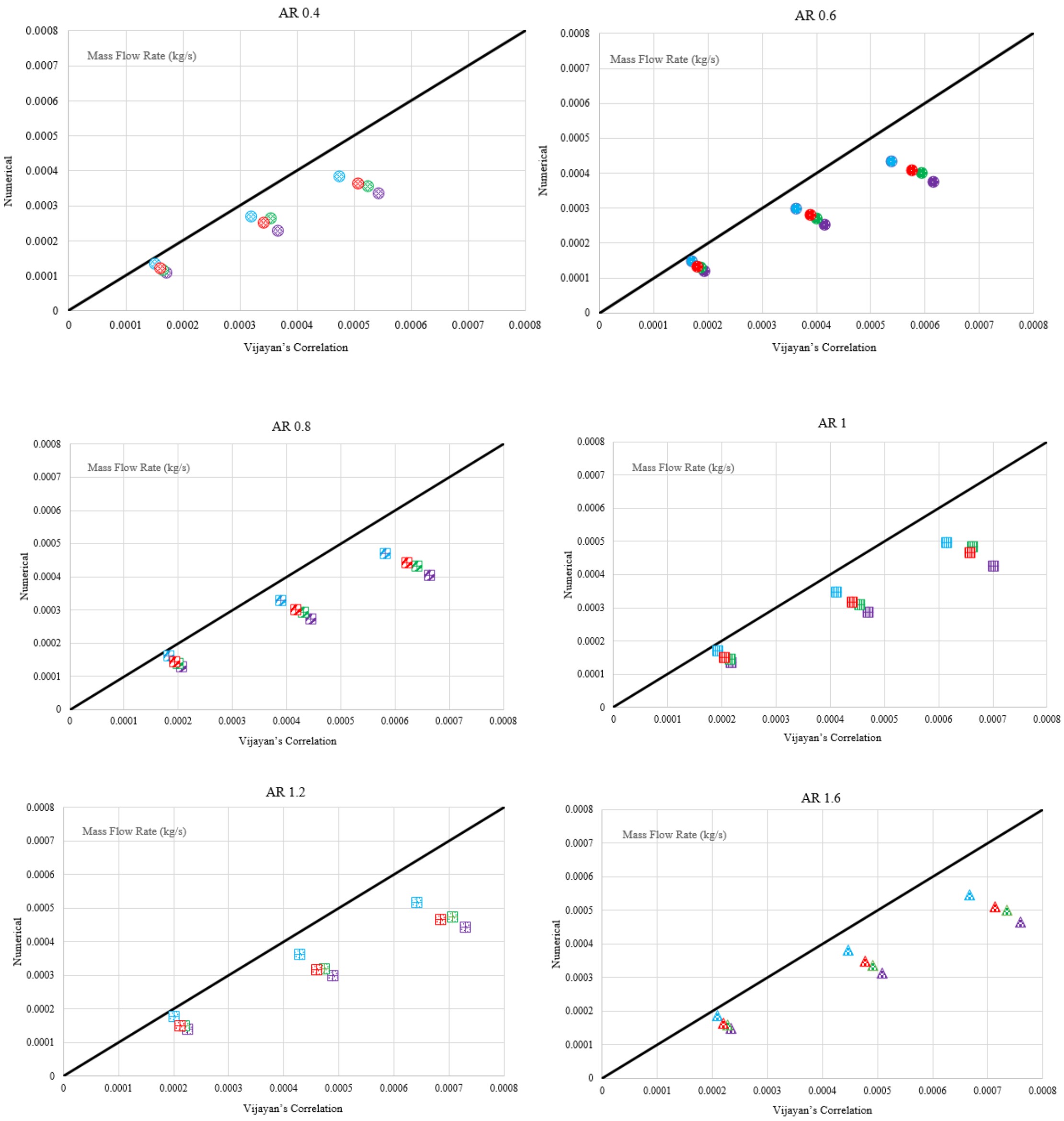


Figure 3.5: Comparison of mass flow rate with Vijayan's correlation for AR

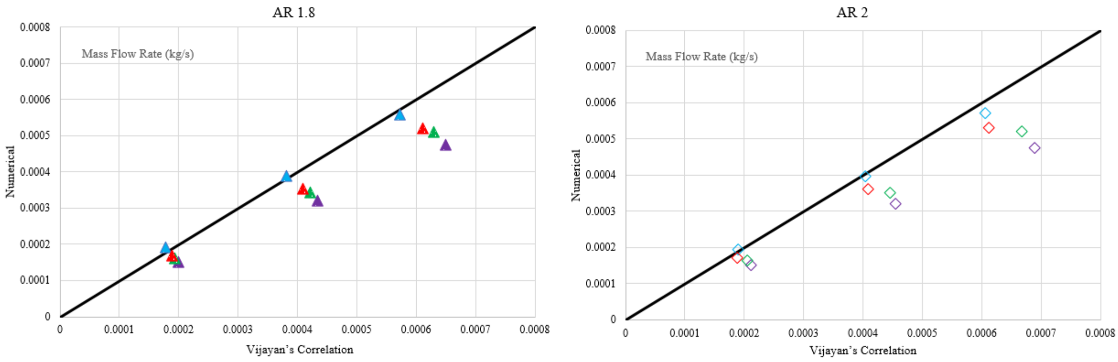


Figure 3.5: (Continue) Comparison of mass flow rate with Vijayan's correlation for AR

3.2.2 Temperature

Increase in heater power and filler content of nanofluid resulted in increase of T_{max} and ΔT_{heater} (Figure 3.6.). The same behaviors have been reported in the literature [14, 15, 18, 20]. However, Bejjam and Kumar [27] found a reduction in ΔT_{heater} when nanofluids were used and higher concentrations increases ratio of reduction of temperature difference. However, ΔT_{heater} increases up to 5%vol. concentration with the increase in heater power, but at 6% vol. it decreased [24].

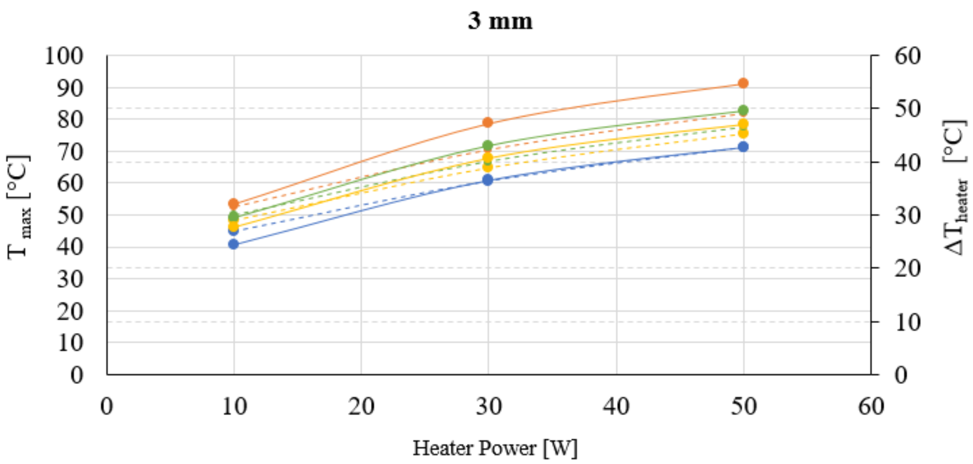


Figure 3.6: T_{max} and ΔT_{heater} for different pipe diameters. (– and -- indicate ΔT_{heater} and T_{max}, respectively.)

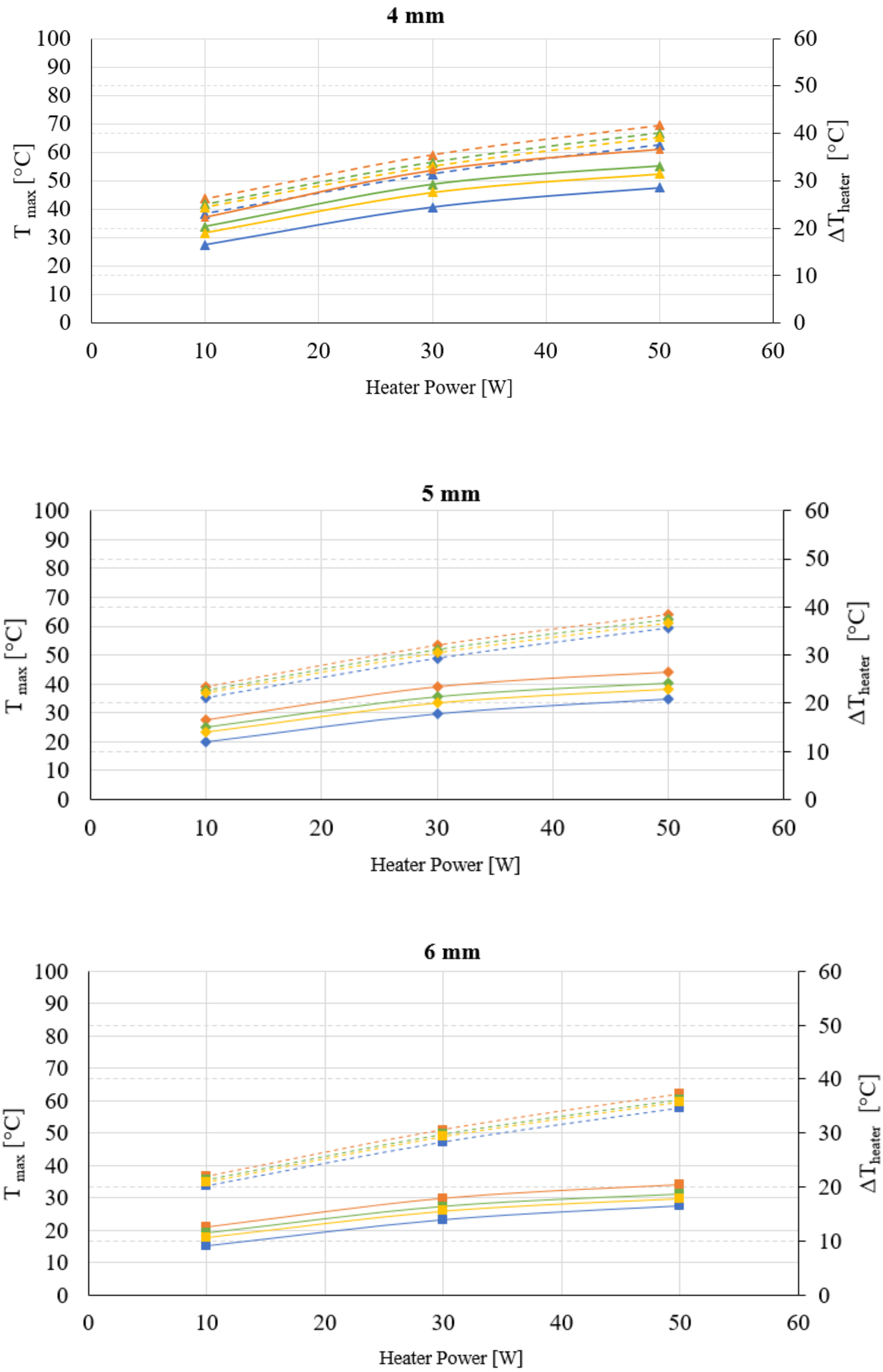
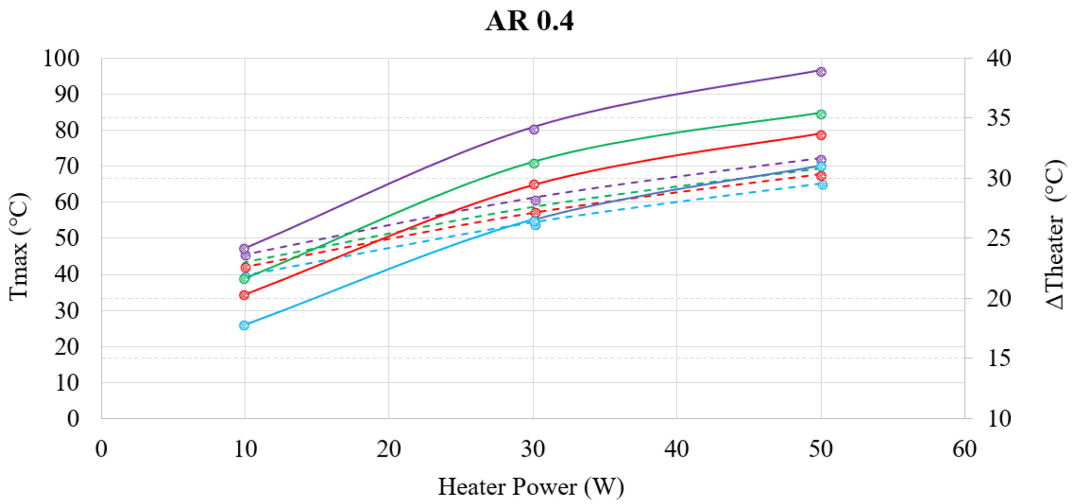


Figure 3.6: (Continue) T_{max} and ΔT_{heater} for different pipe diameters. (− and -- indicate ΔT_{heater} and T_{max}, respectively.)

Similar to pipe diameter Figure 3.7, increase in AR decreases T_{max} and ΔT_{heater}. Nanofluids have higher T_{max} and ΔT_{heater} values (proportional to concentration) at all AR.



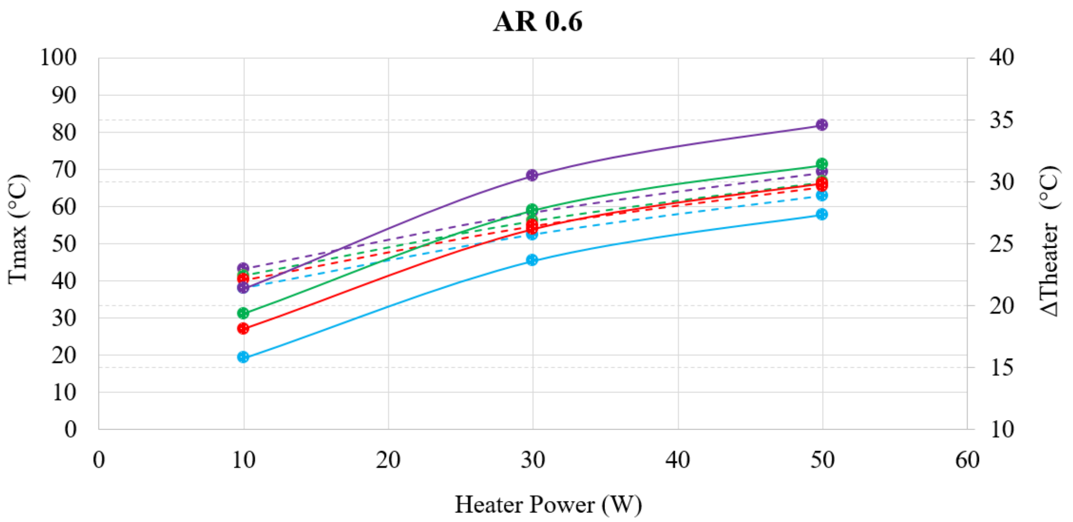


Figure 3.7: T_{max} and ΔT_{heater} for different AR . (– and -- indicate ΔT_{heater} and T_{max}, respectively.)

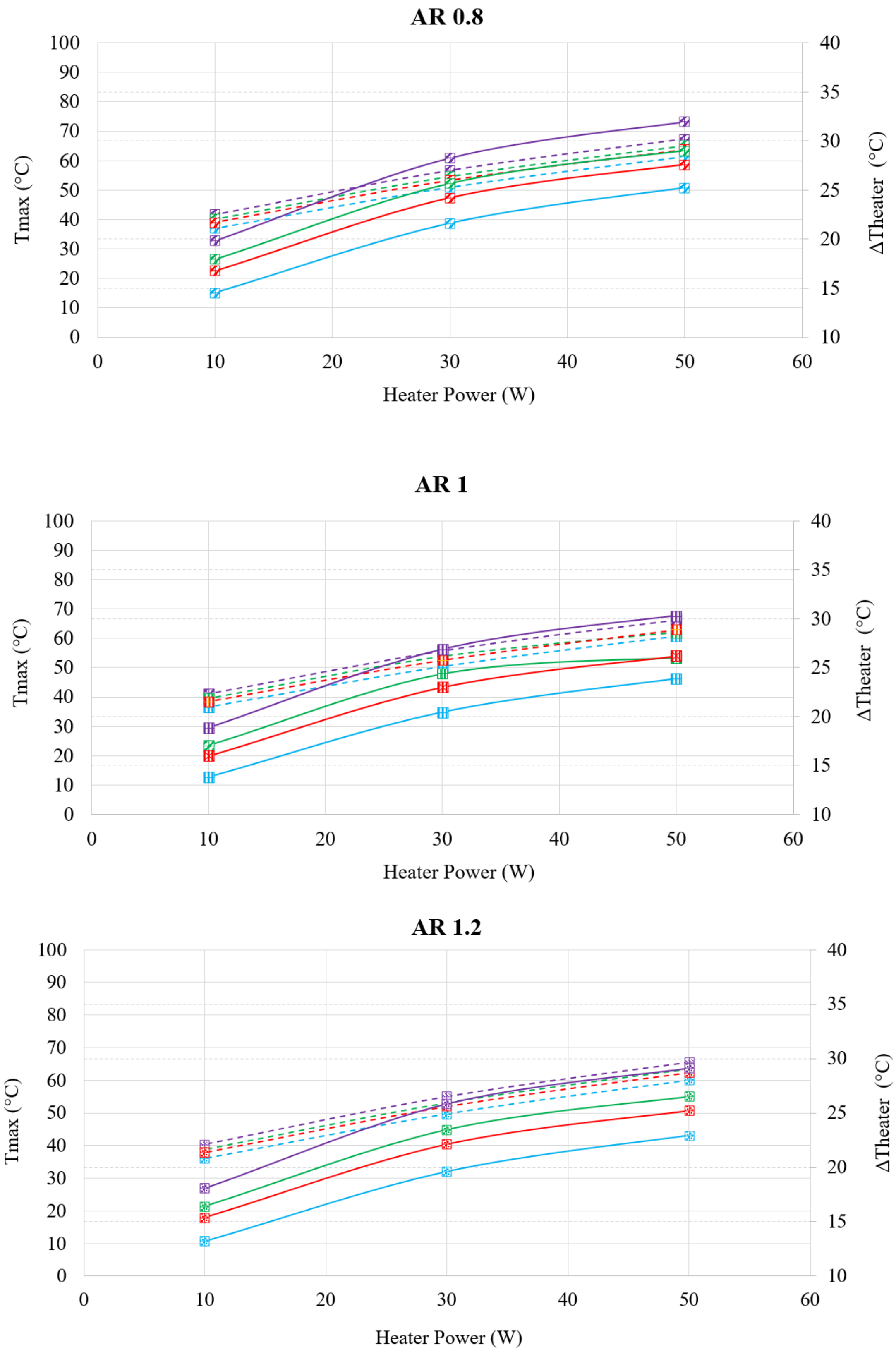


Figure 3.7: (Continue) T_{max} and ΔT_{heater} for different AR . (– and -- indicate ΔT_{heater} and T_{max}, respectively.)

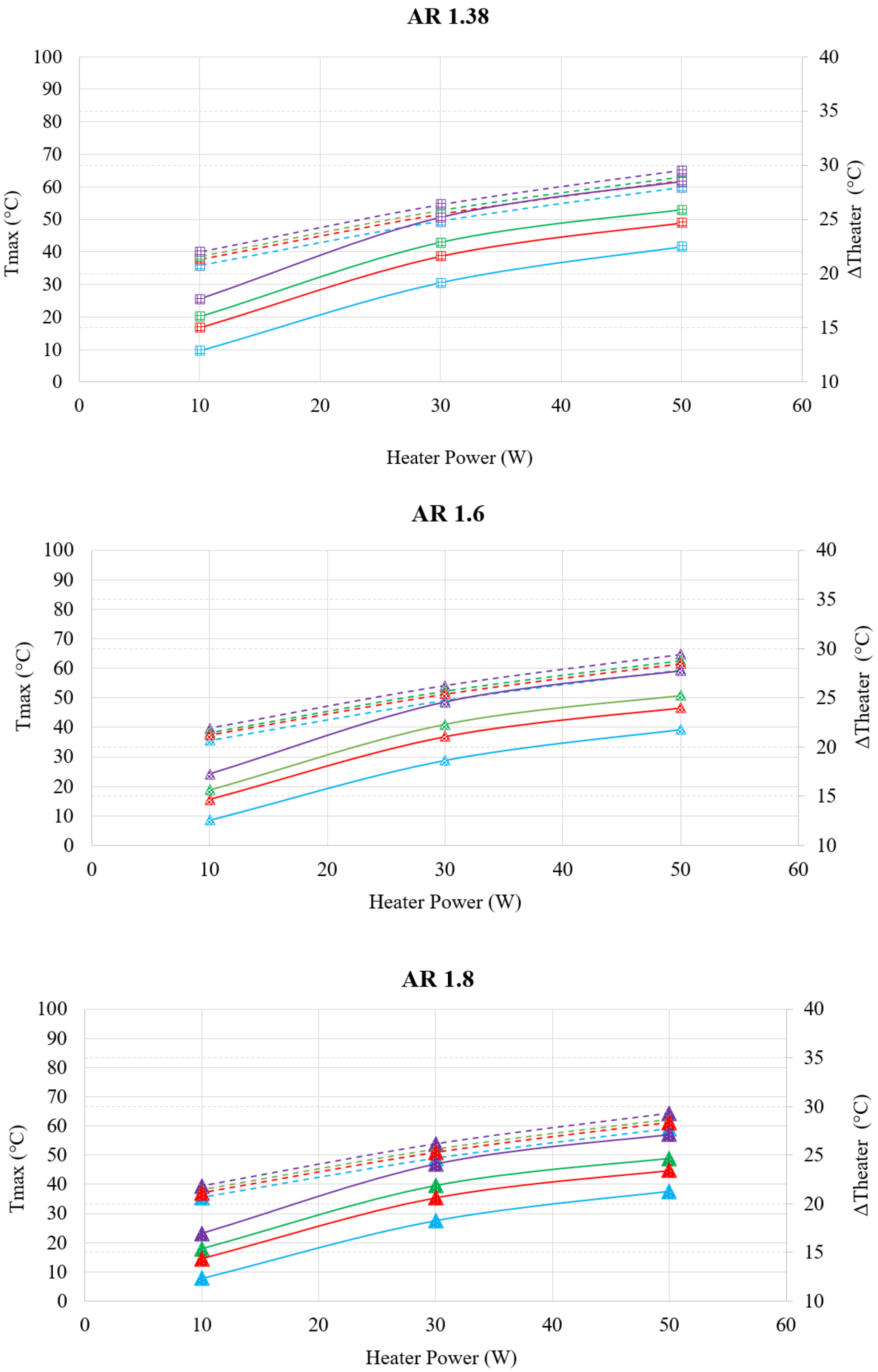


Figure 3.7: (Continue) T_{max} and ΔT_{heater} for different AR . (– and -- indicate ΔT_{heater} and T_{max}, respectively.)

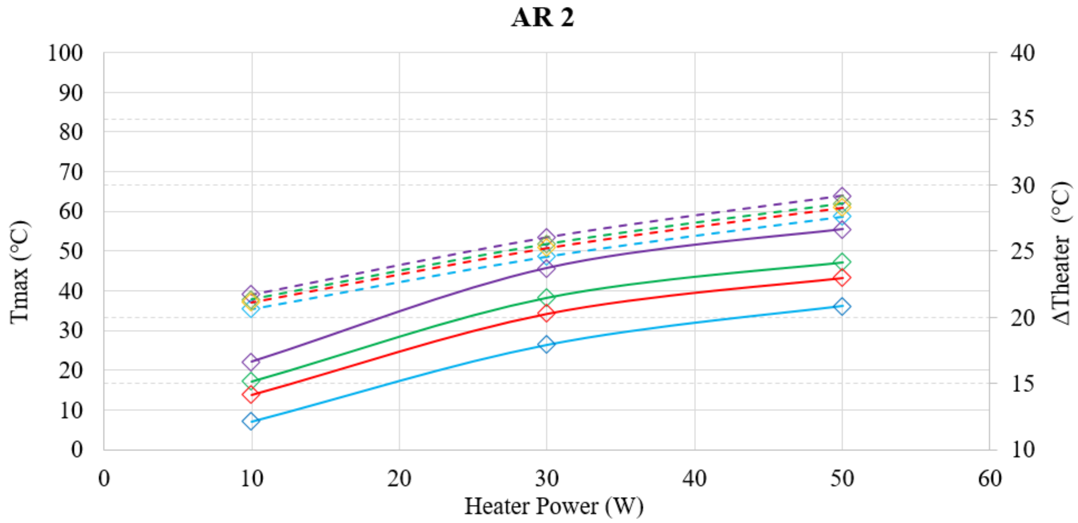


Figure 3.7: (Continue) T_{max} and ΔT_{heater} for different AR . (– and -- indicate ΔT_{heater} and T_{max}, respectively.)

Besides T_{max} and ΔT_{heater}, T_{avg} and ΔT_{avg} are also studied for investigation of thermal performance of the loop for different pipe diameter (3 mm, 4 mm, 4.75 mm, 5 mm, 6mm) in the AR 1.38 and also studied for investigation of thermal performance of the loop for different AR (0.4 , 0.8 , 1 , 1.38 , 1.2 , 1.4 , 1.8 , 2). Figure 3.8-11 present the effects of heater power, nanofluid concentration at different pipe diameters and AR on ΔT_{avg} and T_{avg}. ΔT_{avg} and T_{avg} increases with the increase in filler content of nanofluid and heater power. Although, ΔT_{avg} decreases with the increase of loop diameter, T_{avg} decreases up to 5 mm and then it starts to increase with the increase in heater power. On contrary to these results, Misale et al. [11] has reported that there was no significant difference between nanofluids and DIW for ΔT_{avg} and T_{avg}. For the AR it is seen that T_{avg} increase with increase the heat power and a slight incresing with the low values of AR 0.4. Moreover the ΔT_{avg} has the same reflection with increasing values of the heater but the effect is noticeable when the value of the AR decreases like 0.4.

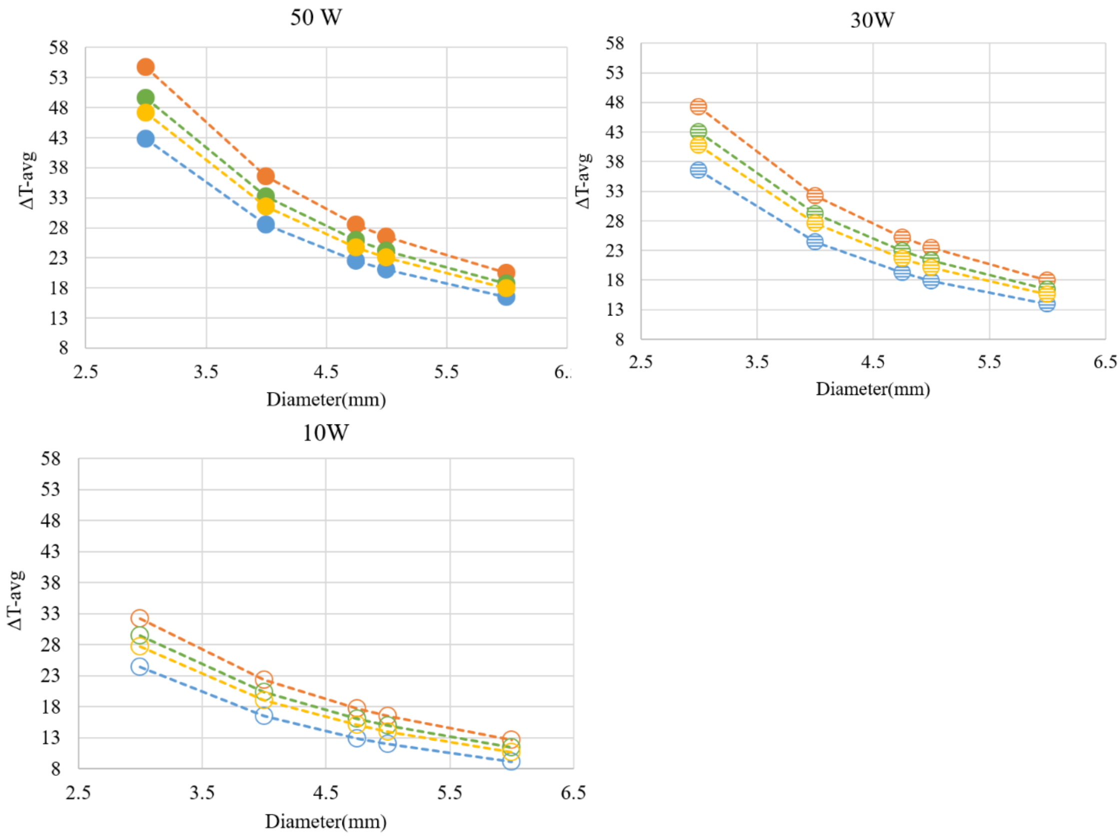


Figure 3.8: ΔT_{avg} at different heater powers with different pipe diameter

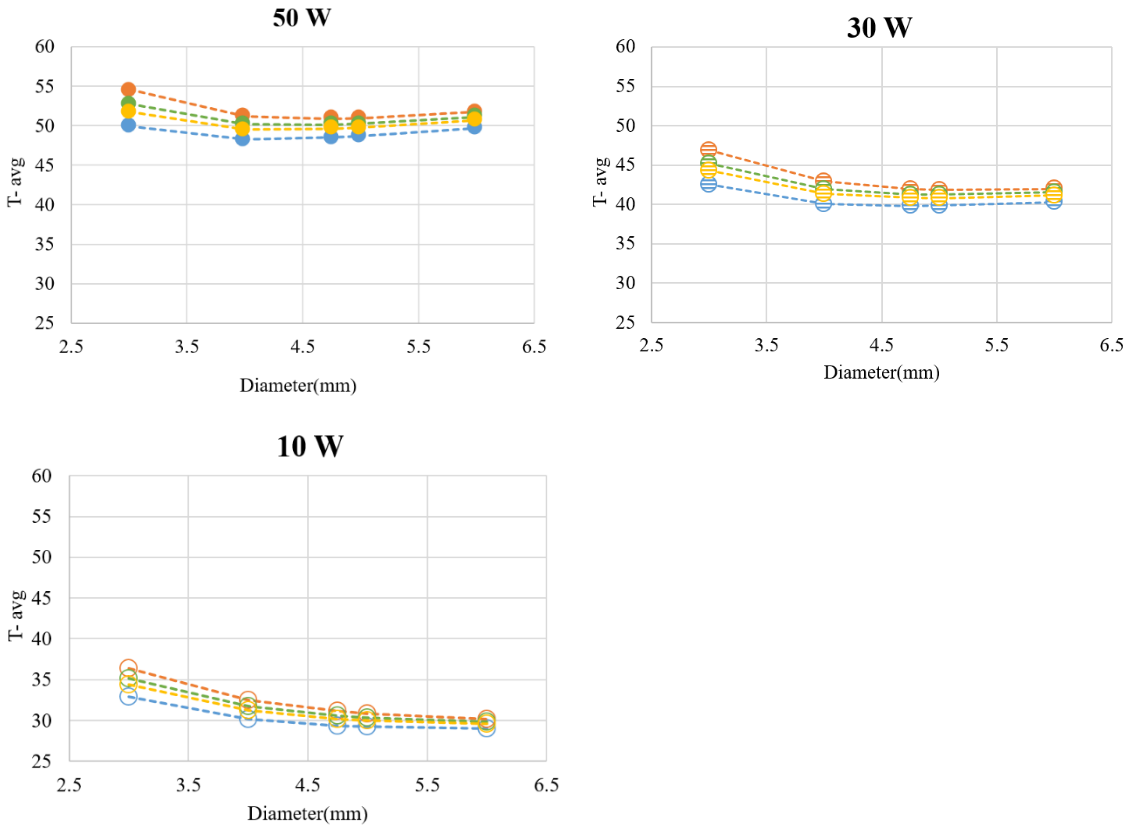


Figure 3.9: T_{avg} at different heater powers with different pipe diameter

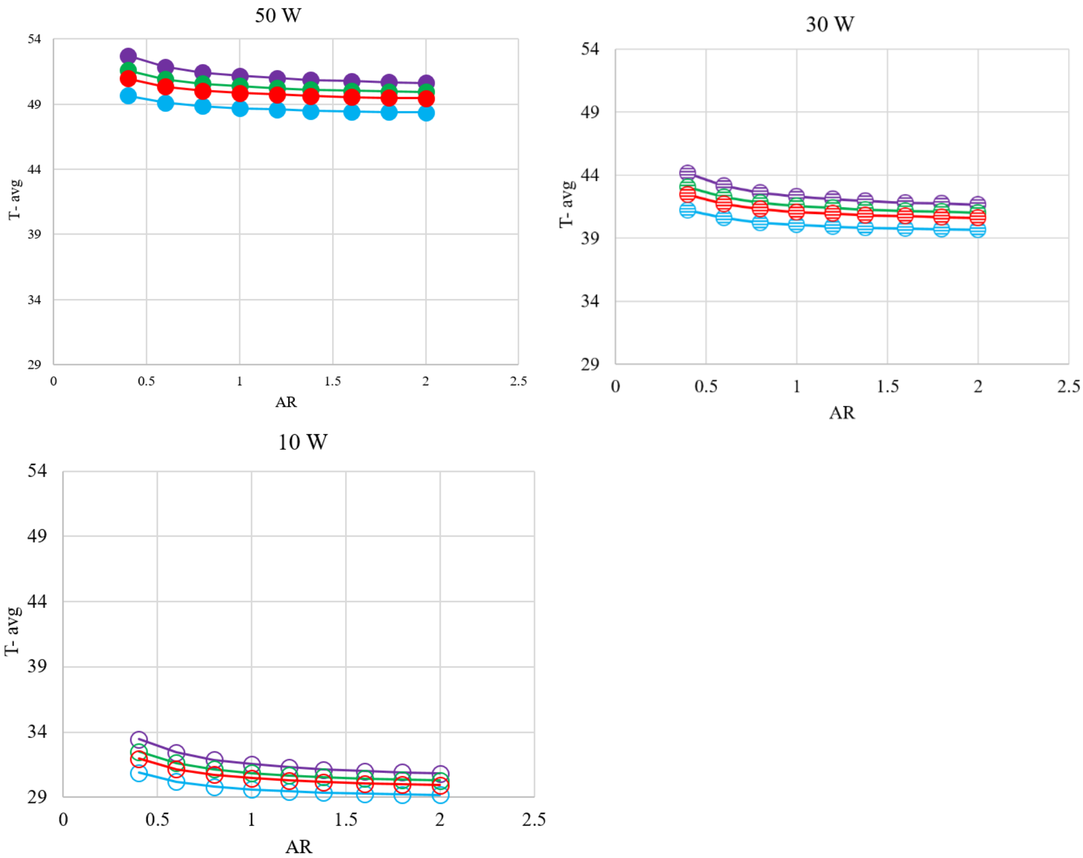


Figure 3.10: T_{avg} at different heater powers with different AR

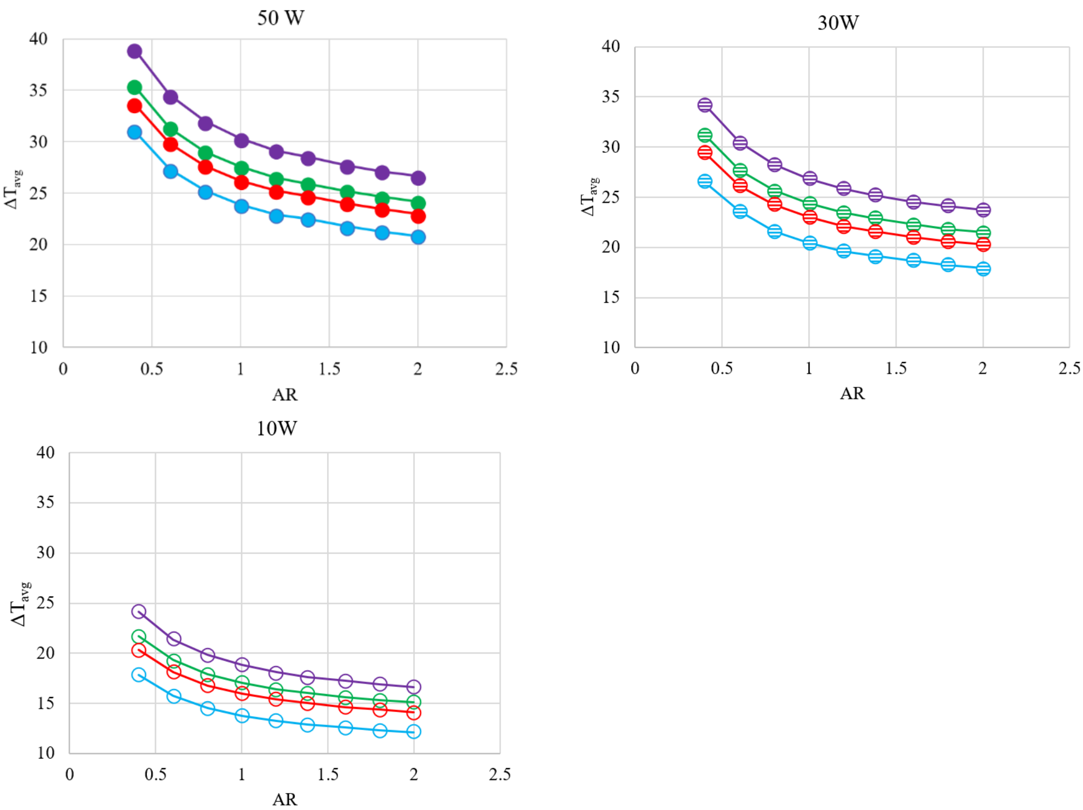


Figure 3.11: ΔT_{avg} at different heater powers with different AR

3.2.3 Effectiveness

Moreover, ε decreases as a result of increase in heater power for all loop diameters as presented The previous studies [15, 16, 18, 19] in the literature has been reported that ε increases with the increase in nanofluid filler content and the results in this thesis are in good agreement with the literaure for all diameters. However, at higher pipe diameters ε have smaller value. Also Figure 3.13 shows that increase in AR decreases the ε factor slightly Figure 3.12.

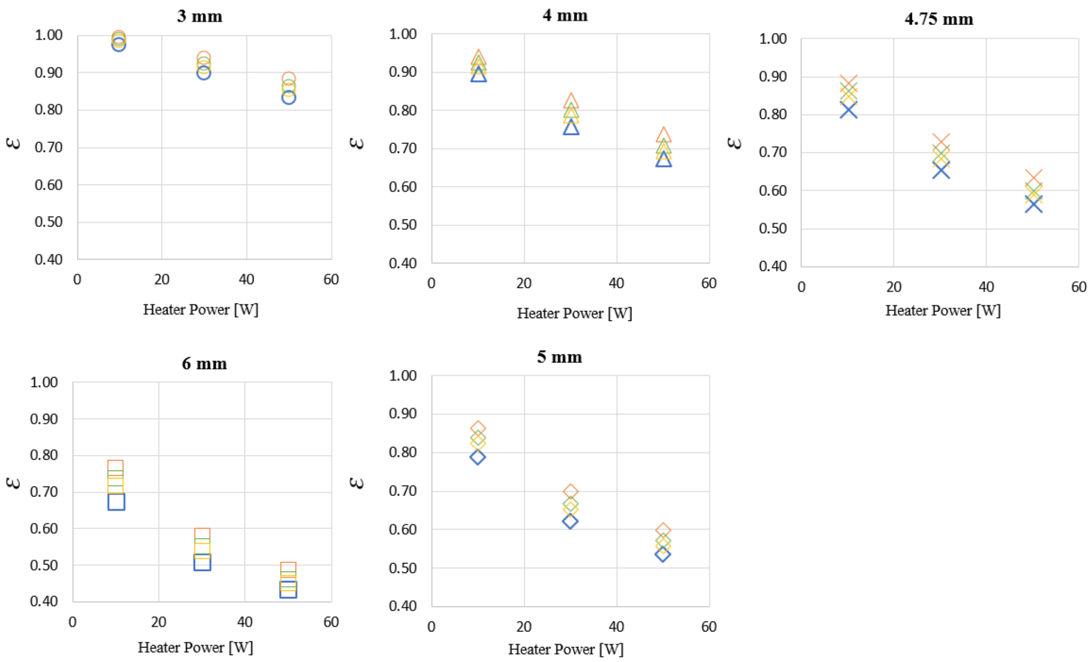


Figure 3.12: ε for different pipe diameters.

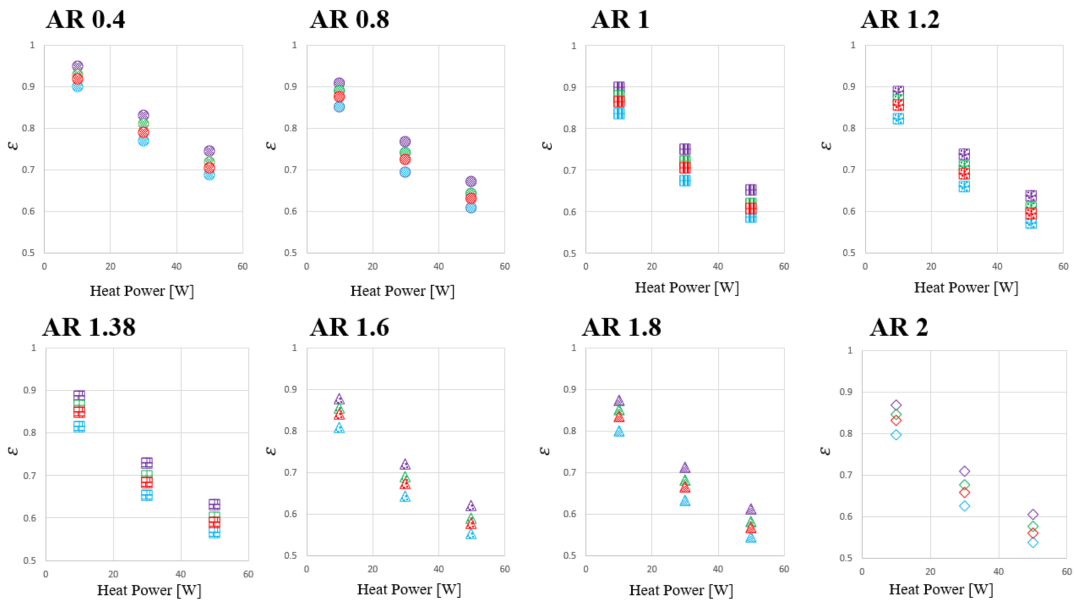


Figure 3.13: ε for different AR.

3.2.4 Overall Heat Transfer Coefficient

The overall heat transfer coefficient U_{in} was evaluated on the basis of the logarithmic mean temperature difference and it is referred to the internal area of the cooler. Overall heat transfer coefficient at the cooler was calculated by using the equations (2.10-13) at lower heater powers and smaller loop diameters, heat transfer coefficient is higher than DIW, however increase in heater power and loop diameter reduces the difference in heat transfer coefficients between nanofluids and DIW (Figure 3.14.). Bejjam and Kumar [27] has explained the reason behind increase of heat transfer coefficient as enhanced thermal conductivity.

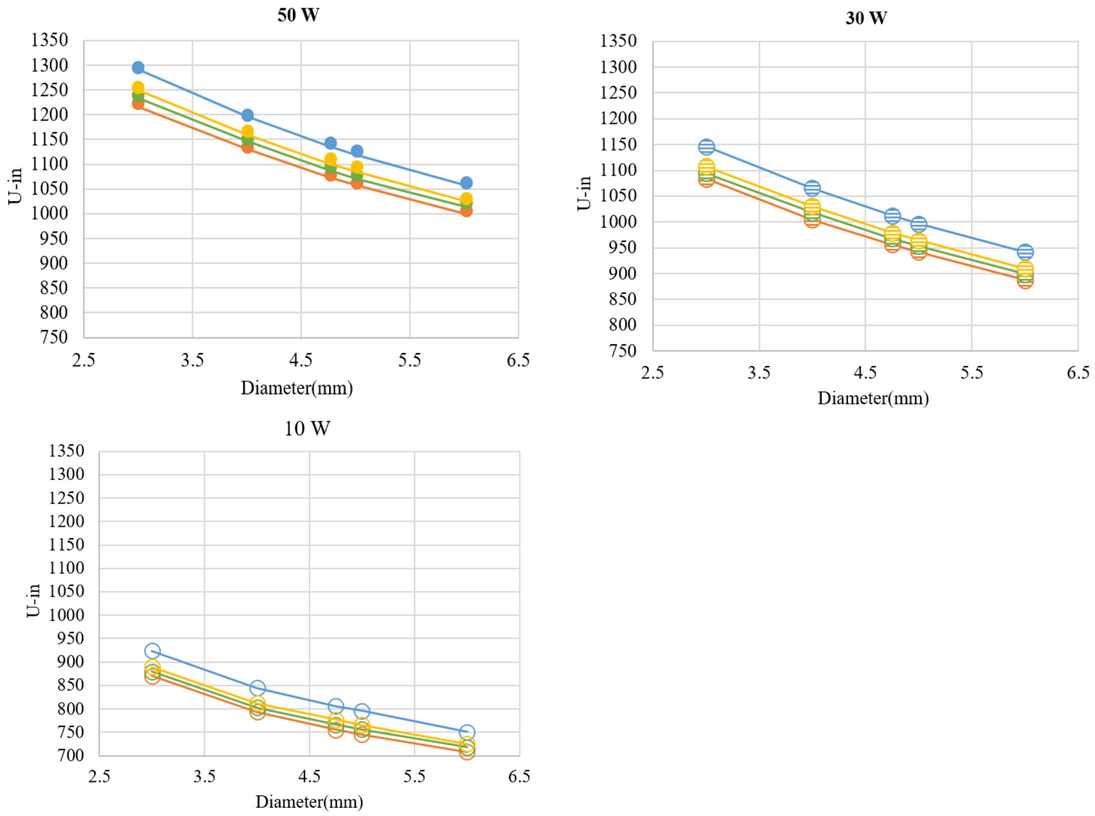


Figure 3.14: U-in for or different pipe diameters.

It is found that an increase of particle concentrations decrease heat transfer coefficient but an increase of AR did not affect significantly.

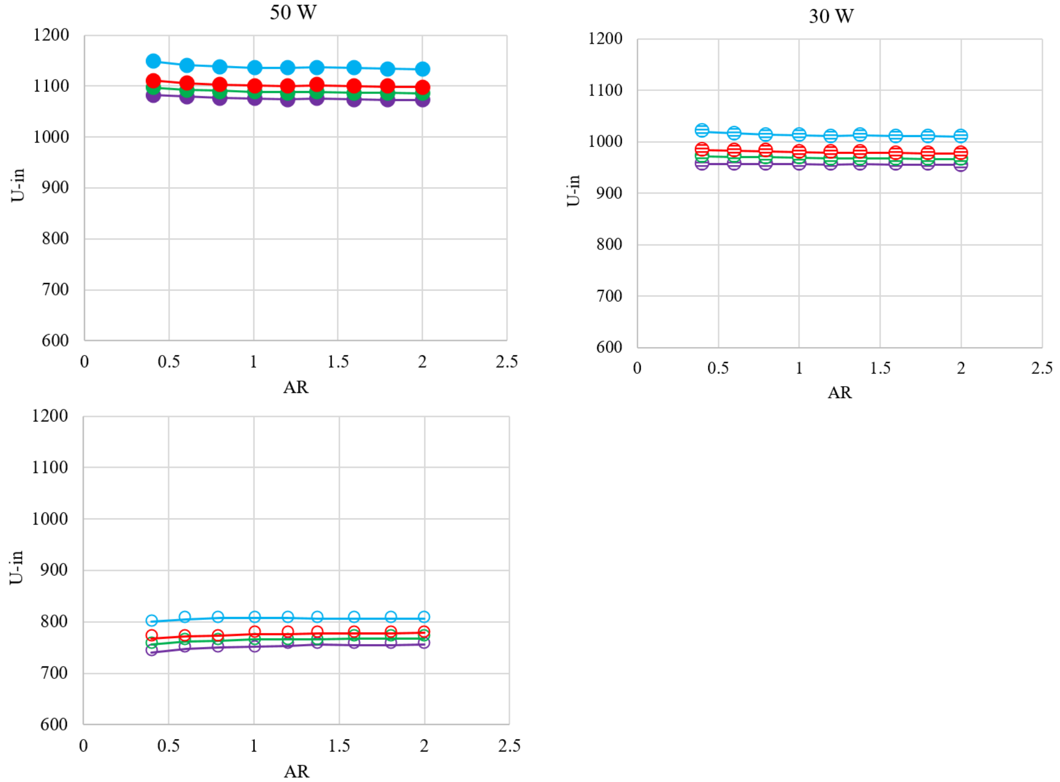


Figure 3.15: U-in for or different AR.

3.3 Temperature Contour

Temperature distributions for the nanofluid and DIW at the critical cross-sections of the loop for different pipe diameters, AR and heater power, in the following figures (Figure 3.16-19). The temperature distributions at the critical cross-sections for the minimum and maximum pipe diameter (3mm and 6mm) that we study, for the same AR of the experimental study (AR=1.38). However, there are local nonuniformities at the cooler inlet, and to approve the previous results the increase of average temperature with increasing heater power and have the highest temperature in the smaller pipe diameter as seen in the difference between Figure 3.15 and Figure 3.17. In the same way, AR for constant pipe diameter (4.75 mm) was studied to show the difference between minimum and maximum values of AR. Contours in Figure 3.18 and Figure 3.19 approve the previous results and show that the average of the temperature will increase for smaller AR 0.4 comparing with AR 2. Moreover, the temperature increases with increasing of the heater power . The same trend has been reported in [27]. Bejjam and Kumar K. [27] observed no temperature variation at the downcomer and riser due to the adiabatic boundary conditions. But in heater and cooler, the upper section has higher temperature contrary to our results.

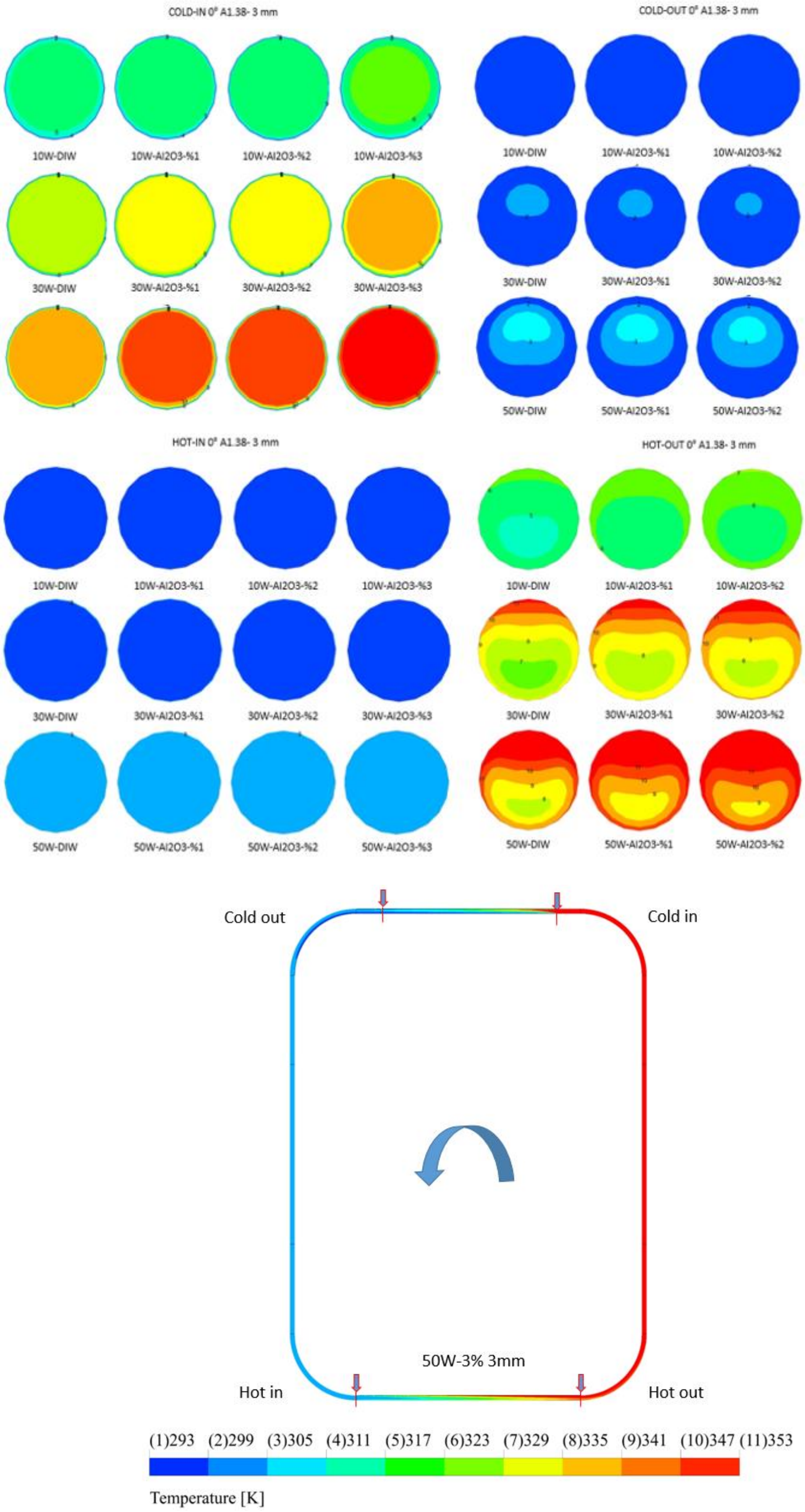


Figure3.16: Temperature Distribution for 3 mm pipe at AR=1.38

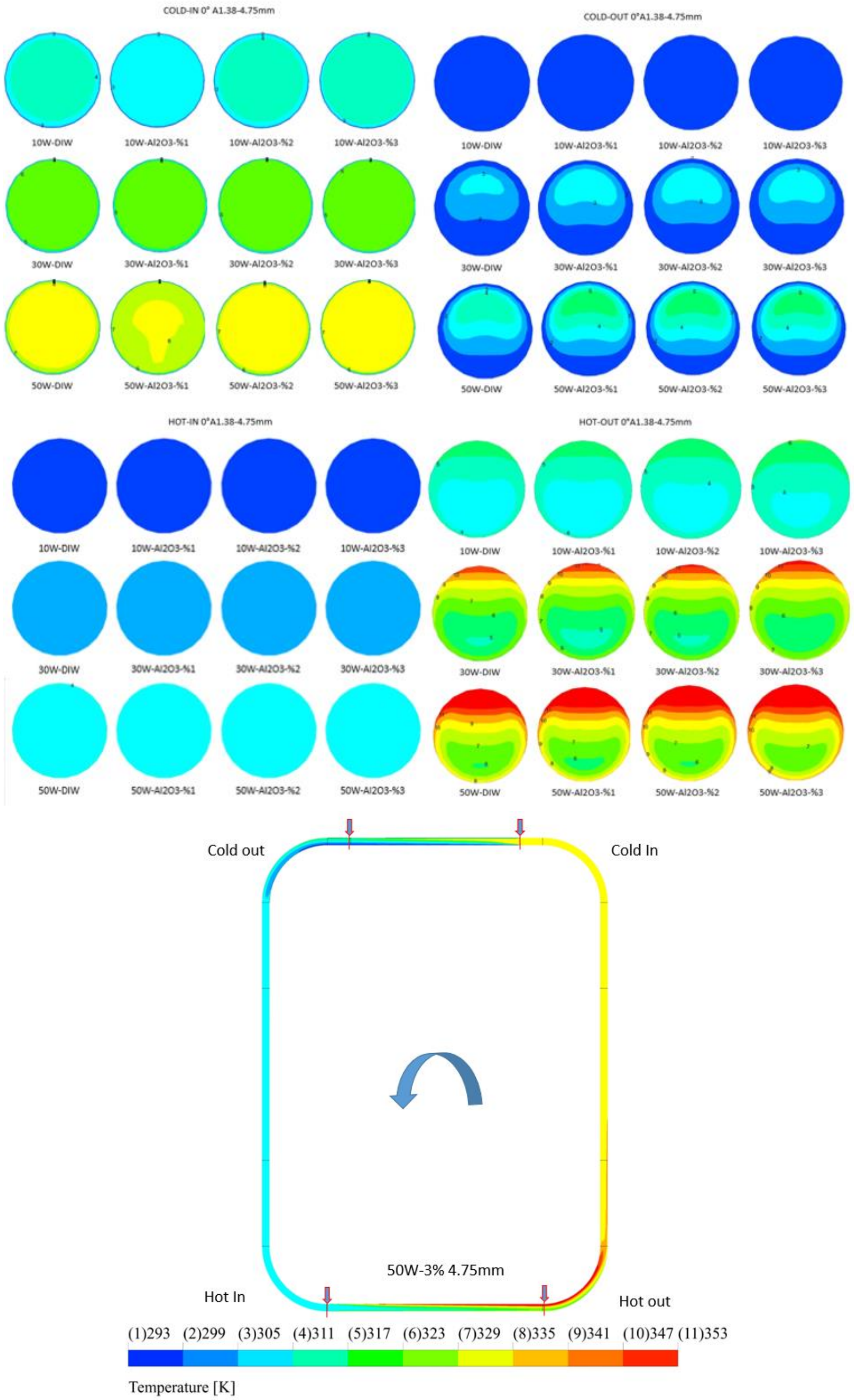


Figure 3.17: Temperature Distribution for 4.75 mm pipe at AR=1.38

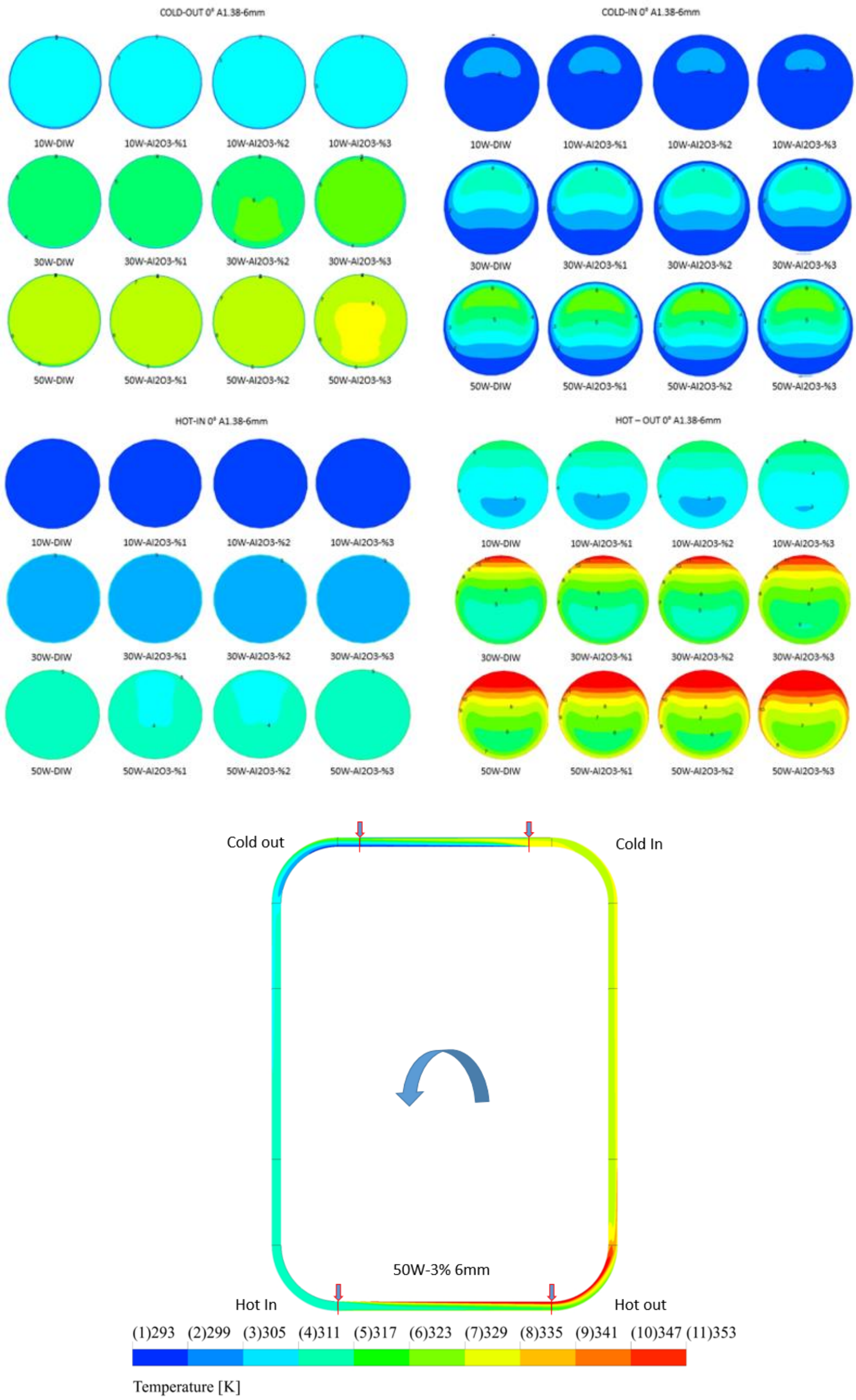


Figure 3.18: Temperature Distribution for 6 mm pipe at AR=1.38

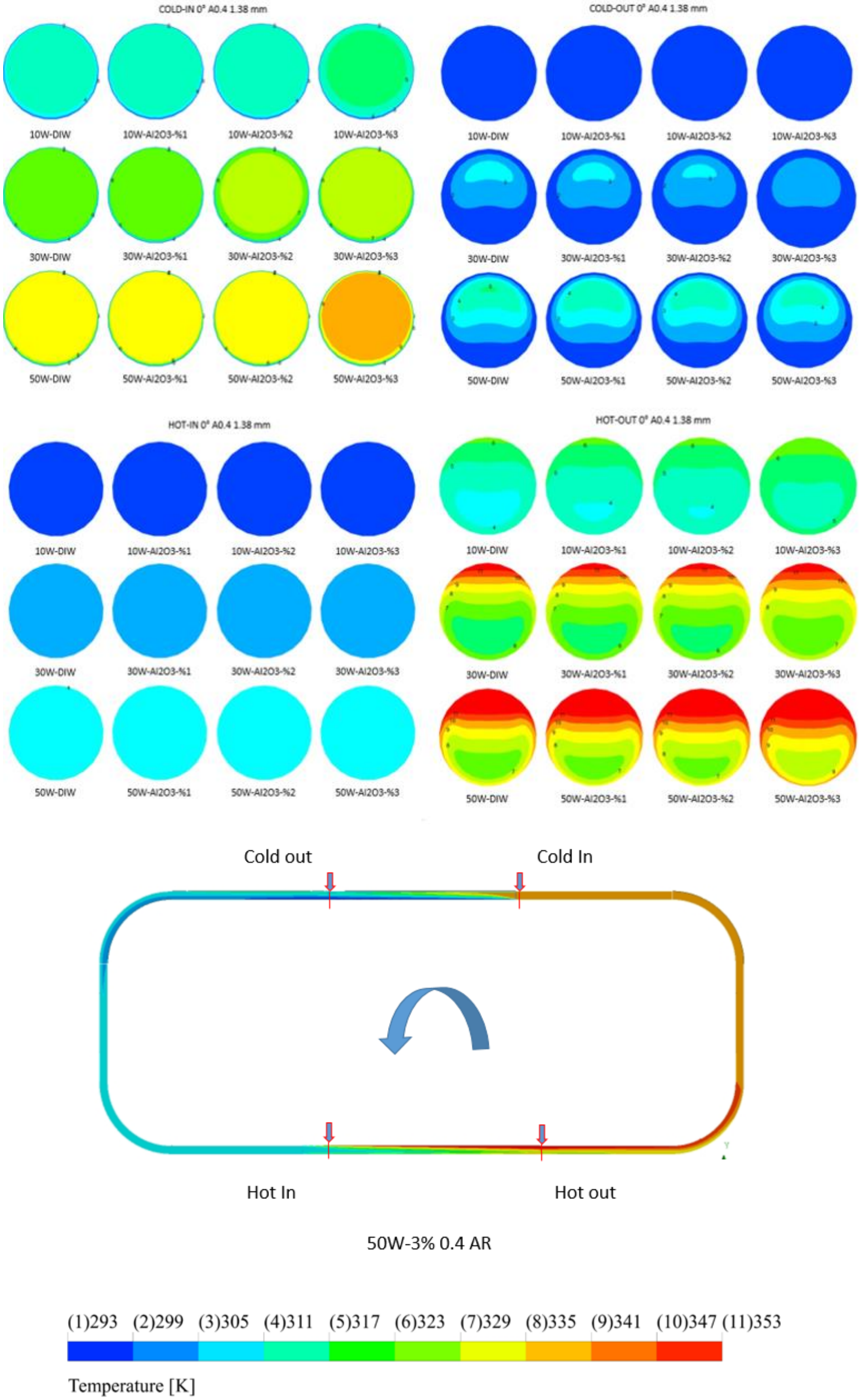


Figure 3.19: Temperature Distribution for 4.75 mm pipe at AR=0.4

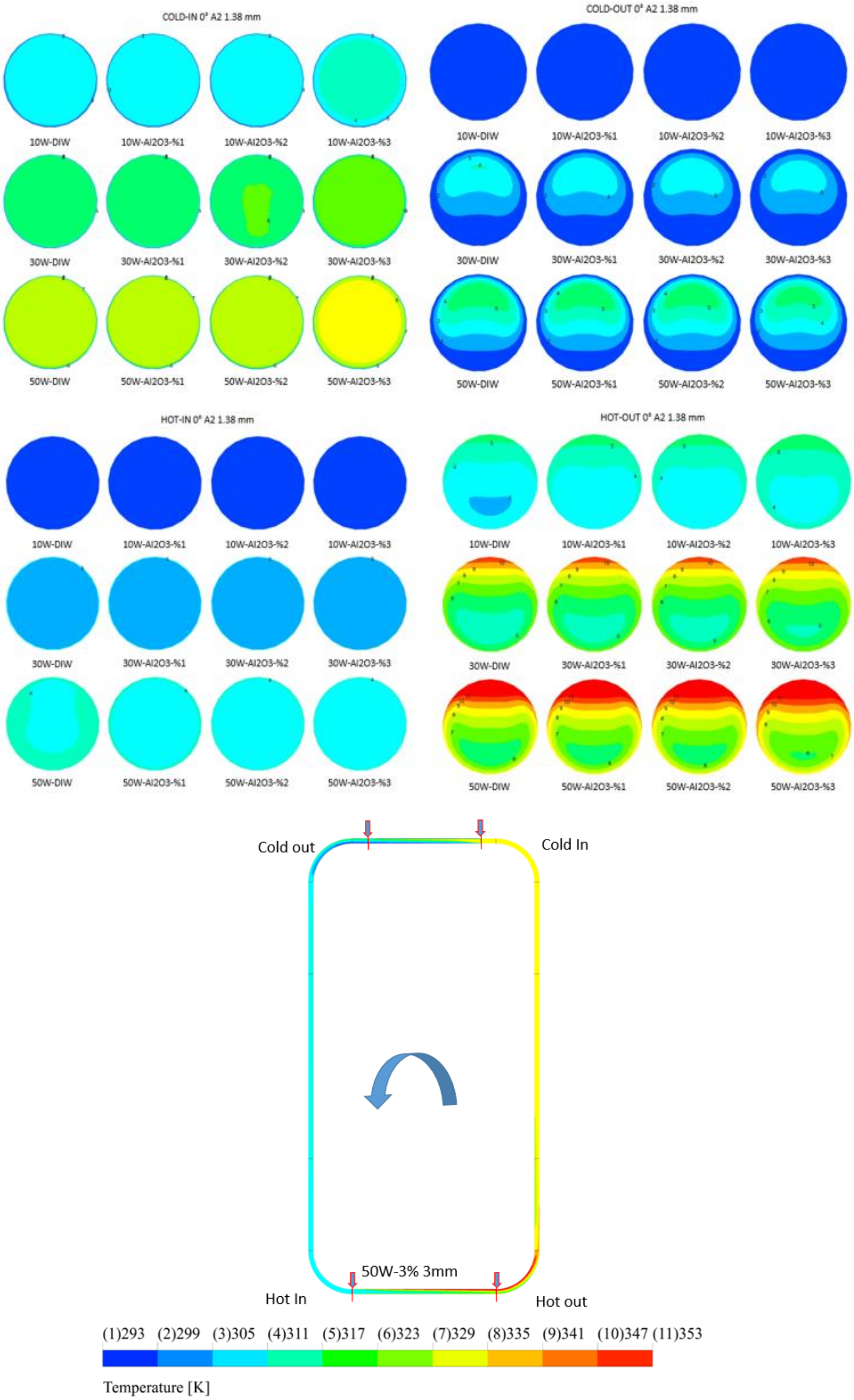


Figure 3.20: Temperature Distribution for 4.75 mm pipe at AR=2.0

4. CONCLUSIONS

In this study, effect of pipe diameter, AR, heating power and volumetric concentrations of water based Al_2O_3 nanofluid on thermo-hydraulic performance of SPNCmL was presented by developed 3D steady state model. The conclusions have been listed as:

- Mass flow rate increases with addition of nanoparticles and for higher heater powers. Up to 4 mm pipe diameter, there is no significant change in mass flow rate. However, for larger diameters increases mass flow rate. Similar to smaller diameters, change in AR does not affect mass flow rate, significantly.
- For same conditions, nanofluids have higher ΔT_{heater} and T_{max} compared to DIW and they increased with the concentration.
- As geometric characteristics, increase of pipe diameter at constant AR and increase of AR at constant pipe diameter were resulted in reduction of ΔT_{heater} and T_{max}.
- T_{avg} and ΔT_{avg} increased with nanoparticle concentration and heater power. Reduction in ΔT_{avg} was observed when pipe diameter and AR increased. T_{avg} was not influenced by change in AR. But for 50 W, up to 4.78 mm T_{avg} decreased and after 4.78 mm it started to increase. For lower heater inputs, up to 4.78 mm T_{avg} decreased and then it was constant.
- Enhanced ε was observed for higher concentration and heater power. Increase in AR and pipe diameter resulted in reduction of ε.
- Nanofluids have lower overall heat transfer coefficient compared to DIW and overall heat transfer coefficient decreased with increase of pipe diameter and heater power. Change in AR did not affect the overall heat transfer coefficient, significantly.
- For heater and cooler inlets have uniform temperature distribution for all pipe diameters and AR. However, temperature varies upper and lower parts of the pipe at the cooler and heater outlets and upper parts have lower temperatures.

- In summary, thermo-hydraulic performance of SPNCmL is affected by geometric characteristics, significantly. This study provides choosing the appropriate geometrical parameters and working fluids for SPNCmLs in desired heat transfer application.

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Office Programs	
Simulation Programs	
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- 3rd International Conference on Thermophysical and Mechanical Properties.
  (EFFECT OF LOOP ASPECT RATIO ON NANO FLUID BASED SINGLE PHASE NATURAL CIRCULATION MINE LOOPS) (SEPTEMBER 2016) Turkey.
- **Engineering Training Program** with Volastic Co.Ltd. Bangkok – Thailand (2014)
- **Graduation Thesis**: Study full HVAC project for national school building in UAE (2013)
- Design and implementation of a vehicle's AC for University lab using a direct Electrical motor (University project) (2011)
- **Supervisor** intern at fabrics production factory in the industrial city of Aleppo(2010)
- **Voluntary** human development with the Syria Trust for Development (2010-2013)

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- Corresponding
- Google AdWords
- SEO

Member Ships

- Member in Engineers Association - Aleppo Branch

References

- Recommendation from Professors in University of Aleppo - Faculty of Mechanical Engineering available uponrequest.

- Assist. Prof. Dr. Ziya Haktan KARADENİZ İzmir Katip Çelebi University (zhaktan.karadeniz@ikc.edu.tr)

Workshop and Courses

- Google AdWords & SEO Course (2017)
- German Language Courses : (Izmir Goethe Institute)
- English Courses : (Higher Institute of Languages)
- Turkish Courses : (level 2 - New Horizon & still)
- Attended courses Project Management in The Syria
- Attended Project Management Software (Primavera)
- Certificate Computer skills of the Syrian Computer
- Document of success in Foreign Language Course
- Document the success of the examination of the